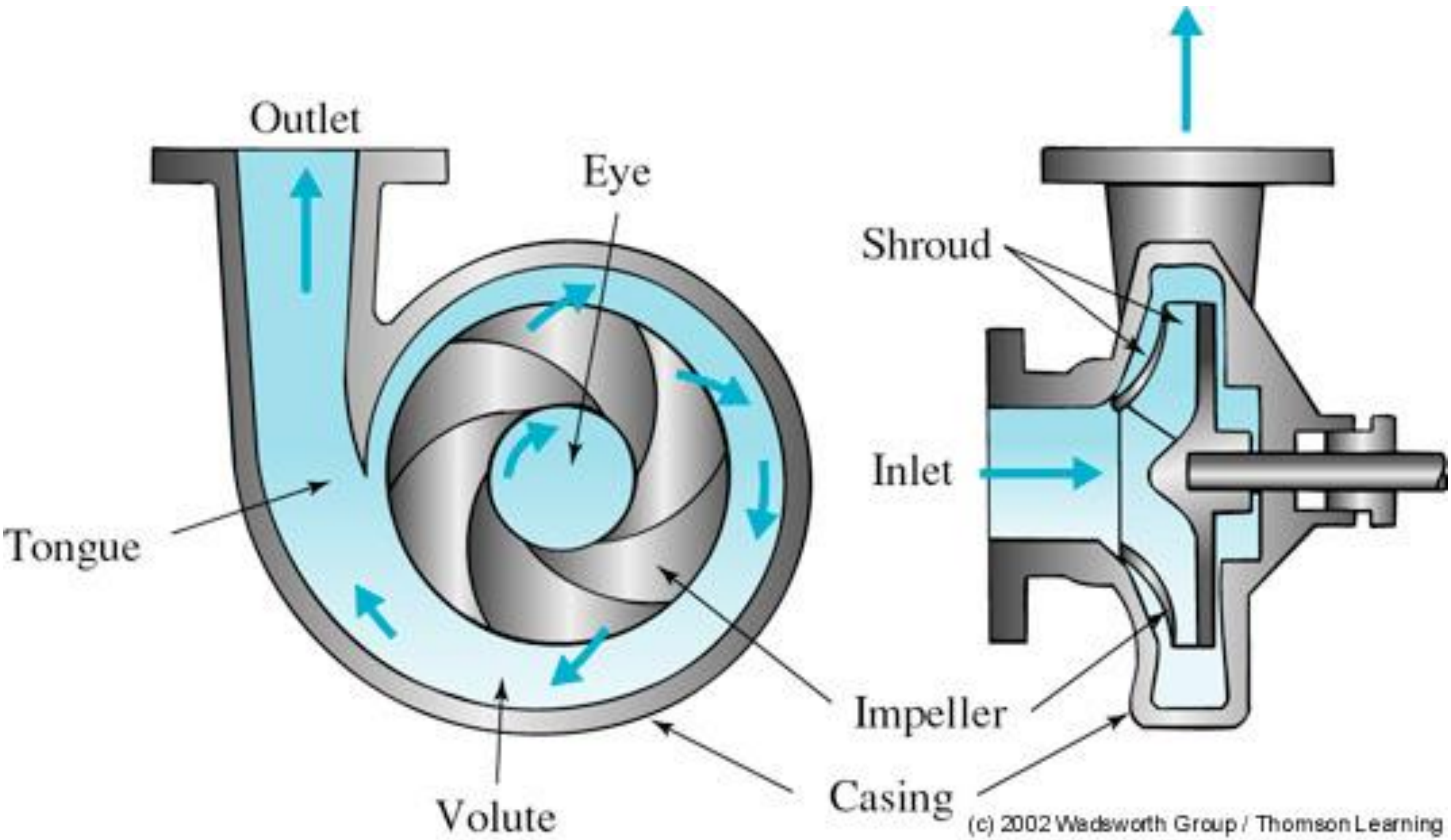
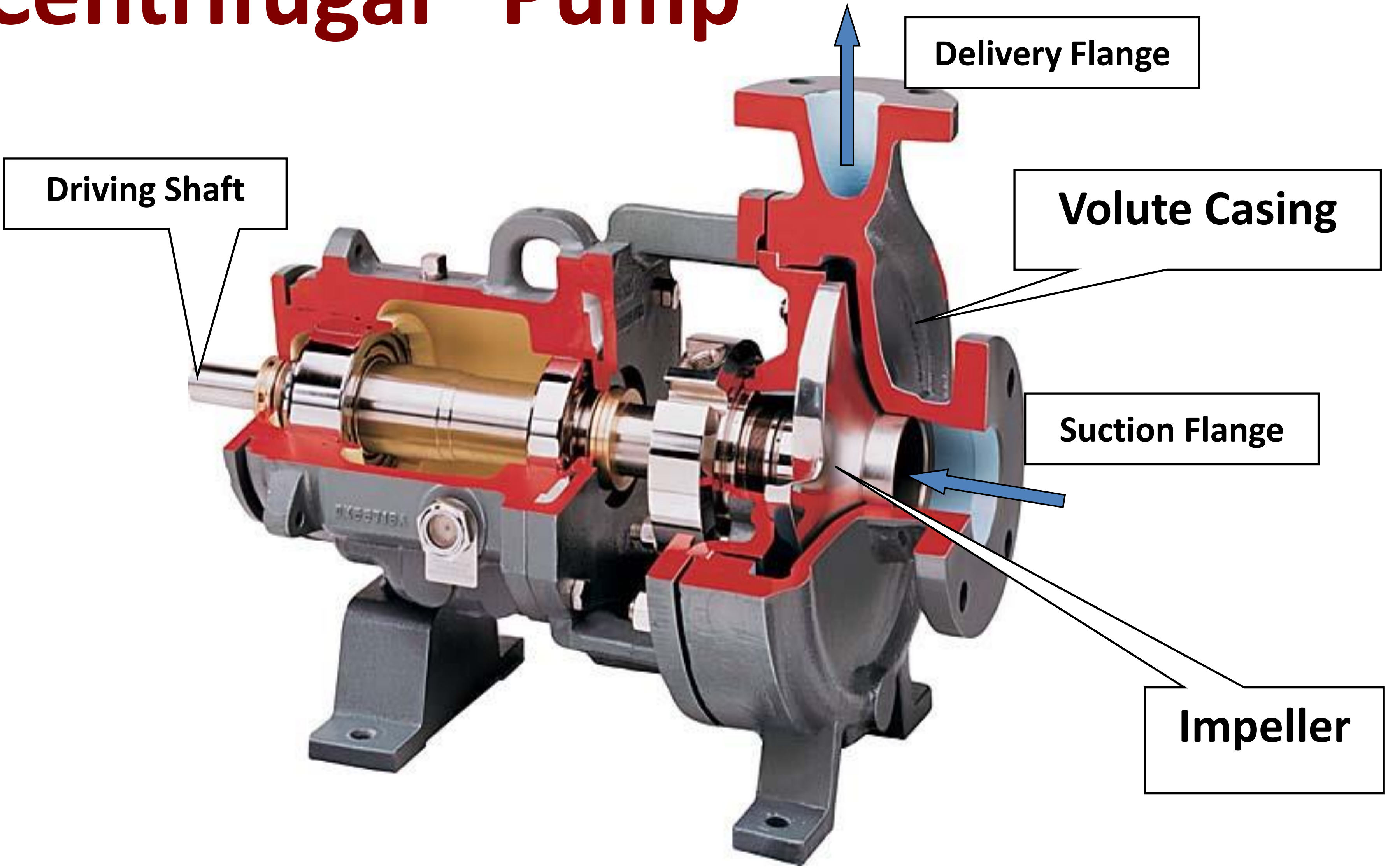
CENTRIFUGAL PUMPS COURSE



Single-suction radial flow pump.

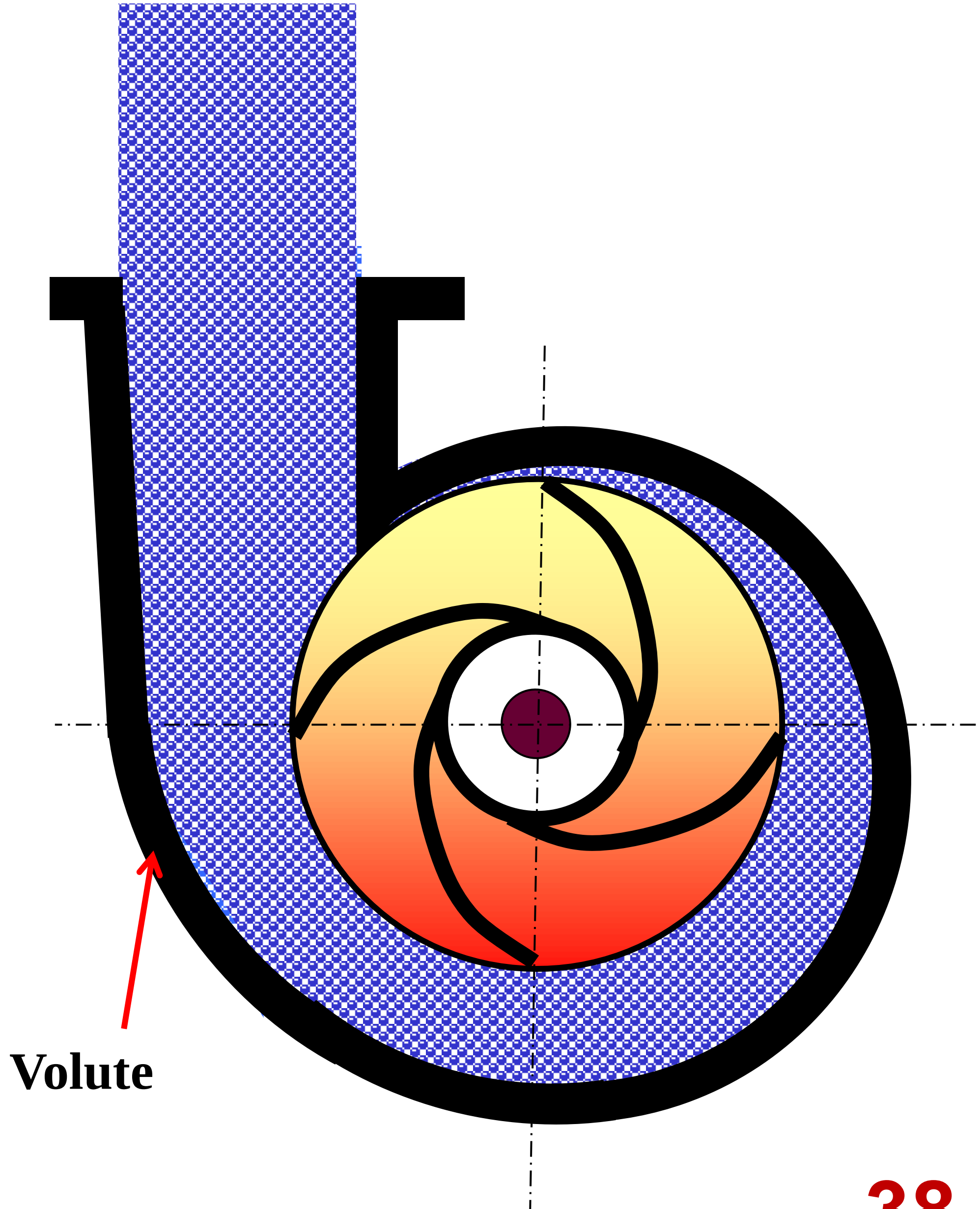
Centrifugal Pump



1- WITHOUT DIFFUSER

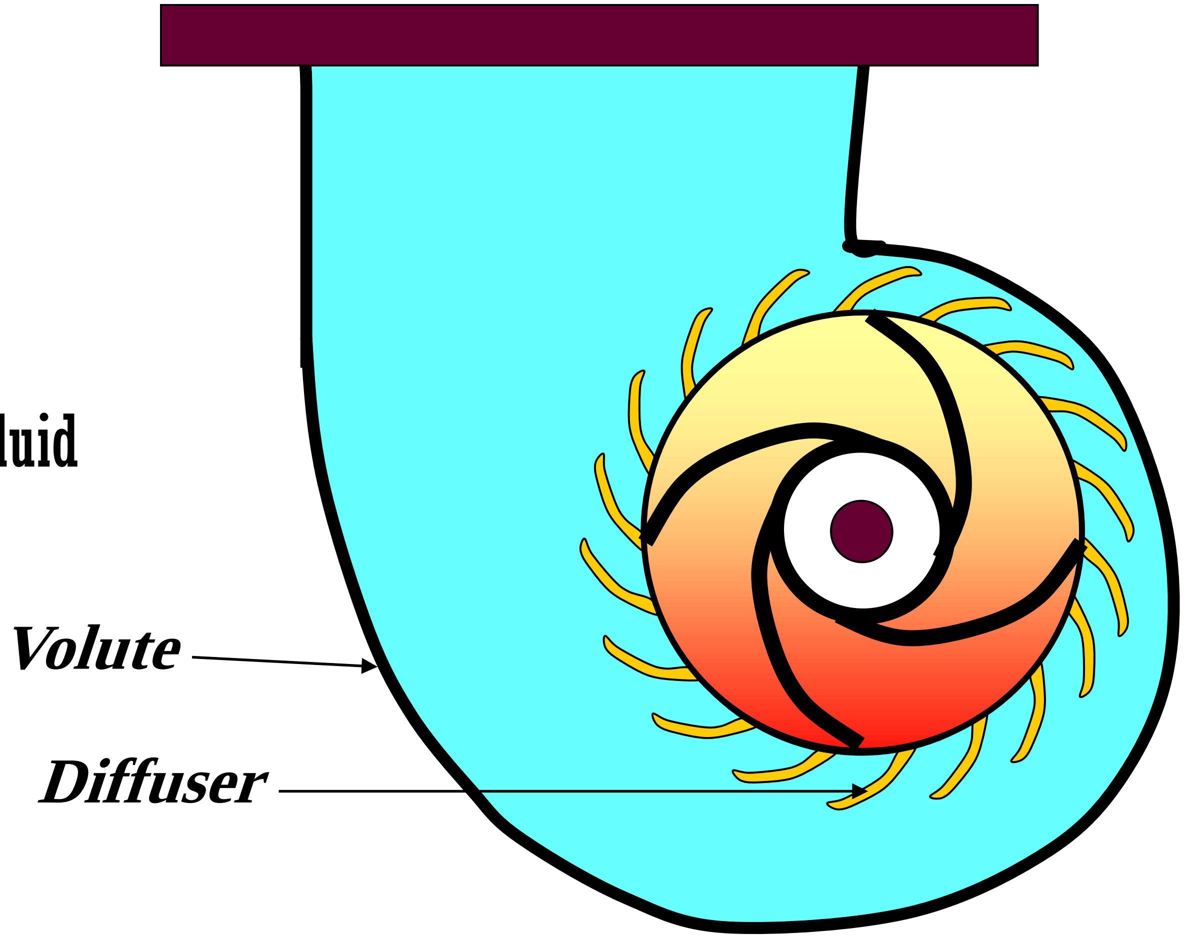
Volute casing is used to convert part of Velocity energy at impeller exit to pressure energy.

$$(U^2/2g)$$

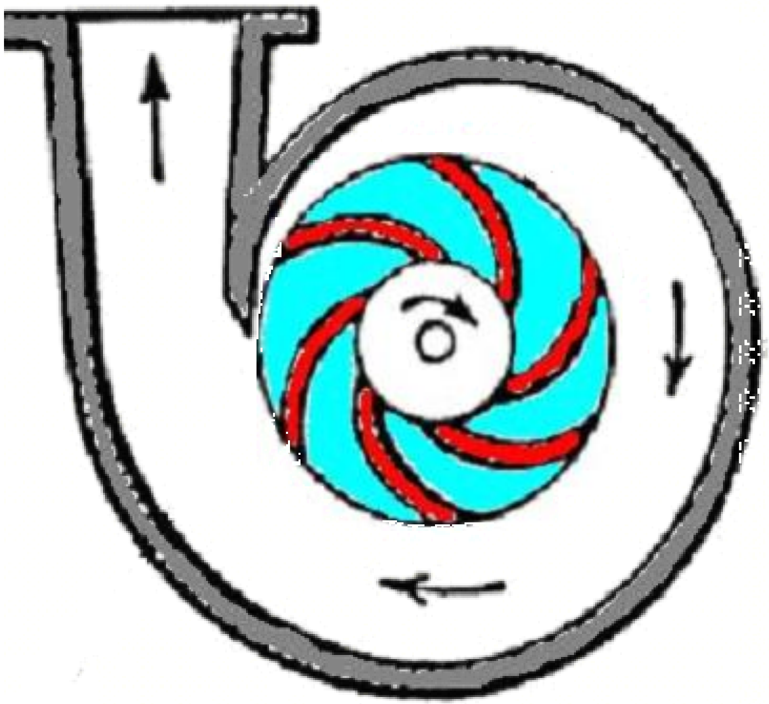


2- WITH DIFFUSER

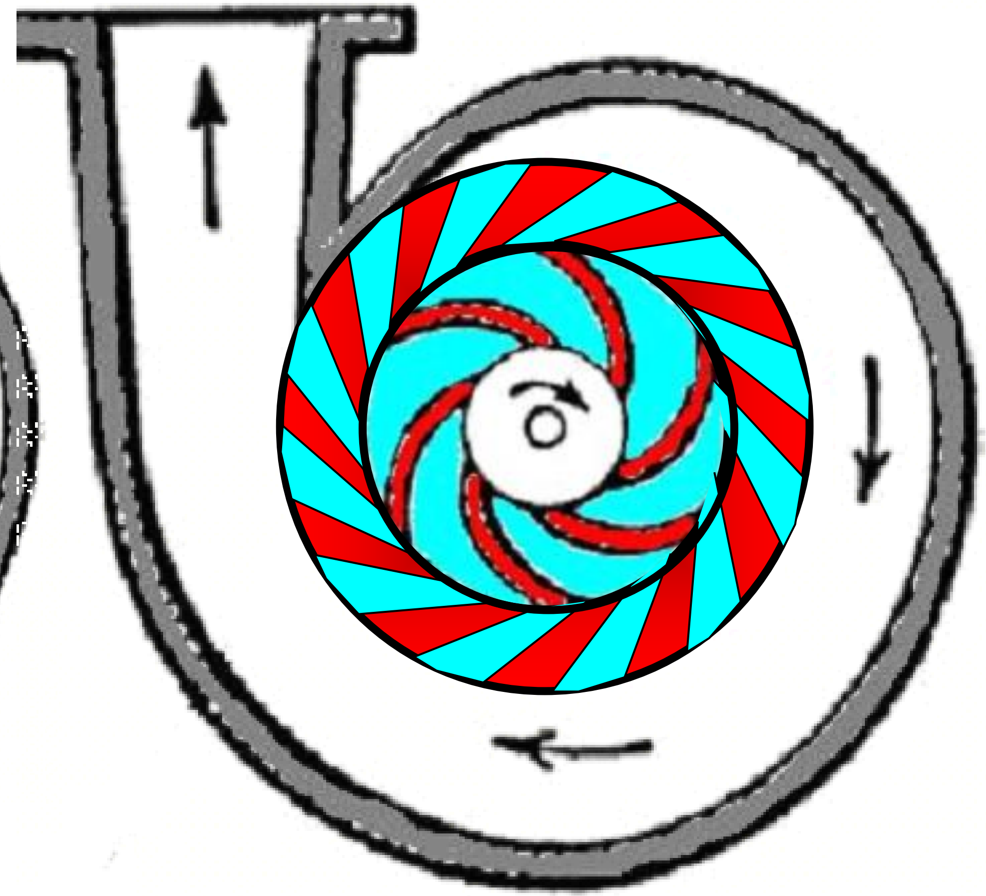
Diffuser function is to decrease the turbulence losses and unify the direction of the outlet fluid



Centrifugal Pump



Volute



Diffuser

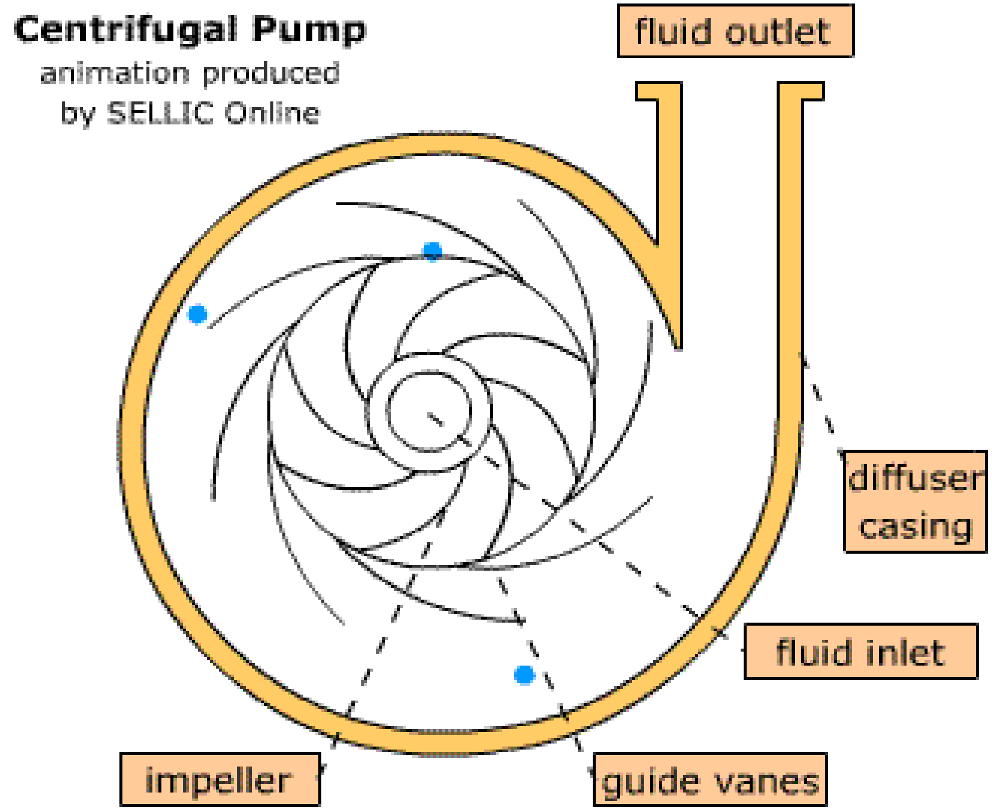
Centrifugal Pumps

Definition:

Centrifugal pumps increase momentum and pressure head by means of rotating blades which converts radial velocity into pressure head.

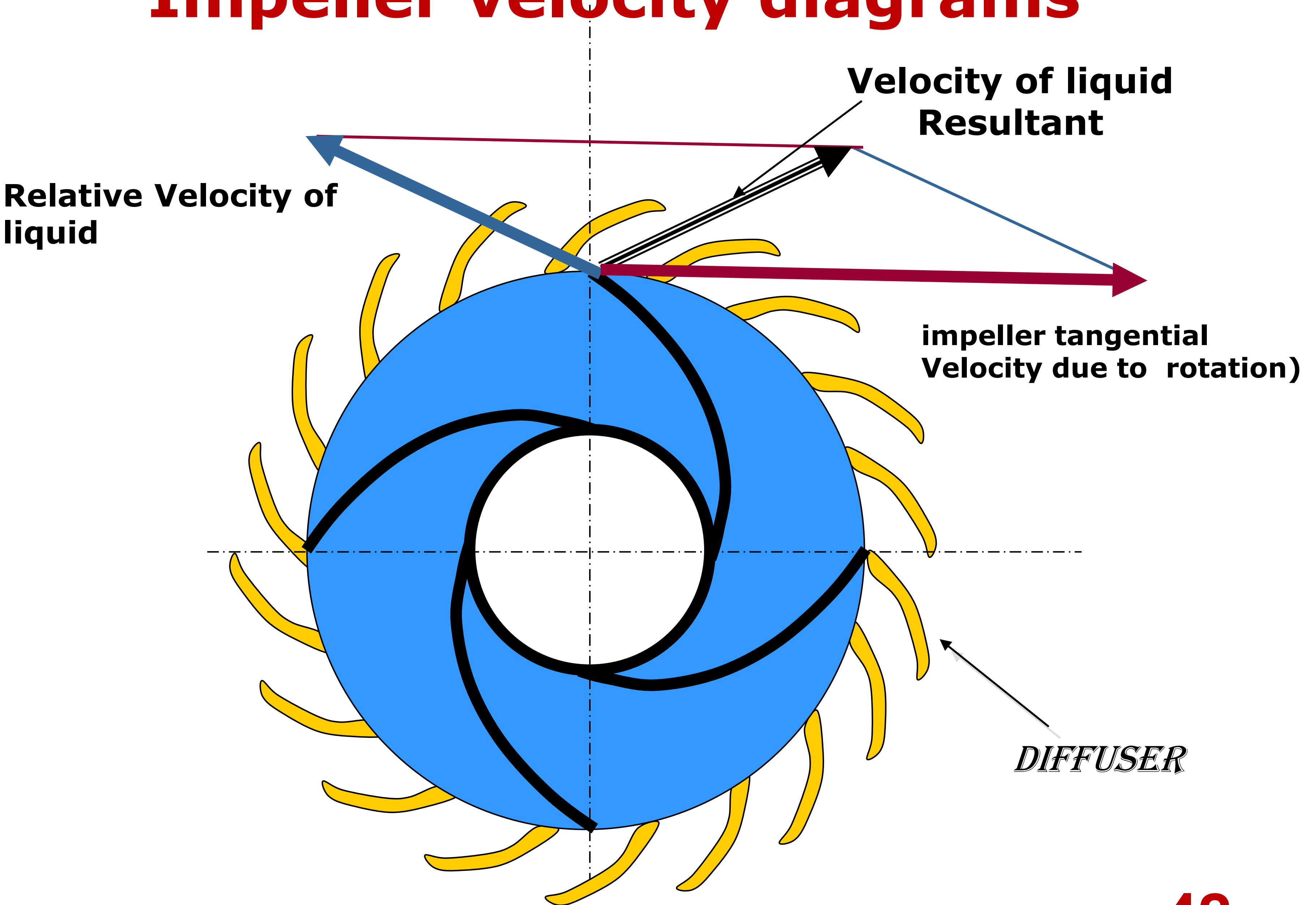
Components

– Inlet duct
– Impeller
– Volute
– Discharge nozzle

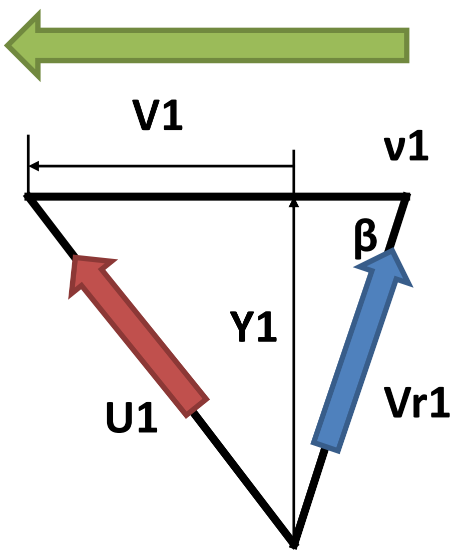


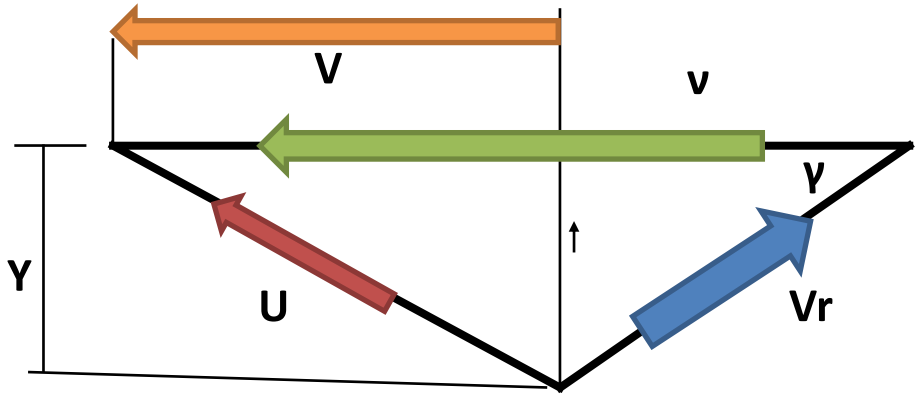
Centrifugal Action of C.P. Impeller

Impeller velocity diagrams



Velocity Triangles of liquid due to impeller rotation





Outlet Vel. Triangle

$$Q = Y\pi DB = Y_1\pi D_1 B_1$$

$$v = \frac{\pi DN}{60}$$

Theory of centrifugal pump impeller

To develop the basic theory of centrifugal pump, it is assumed that :

The impeller consists of infinite number of uniform smooth vanes of zero thickness. Using the following symbols:

T= torque on impeller shaft.

ω= angular velocity of impeller.

H = Ideal head developed by pump impeller.

γ= is the outlet blade angle made by the relative velocity vector with v.

β= is theinlet blade angle made by the relative velocity vector with v_1.

r = radius, B = width of flow passage

Newton's 2nd law stated that; the torque =the rate of change of angular momentum.

$$\text{i.e.} \quad T = d[mV \times r]/dt$$

$$T = m'[V \;.r - V_1 \;.r_1]$$

$$T = \rho Q\,[V\,.r - V_1.r_1]$$

V is the liquids velocity component in tangential direction, i.e. perpendicular to the radius(Whirl Component).

W.D./unit wt =Tω/ ρgQ =

$$\omega[V\,.\,R - V_1.r_1]/g = (V\,v - V_1 v_1)/g$$

For Maximum W.D./unit wt , V_1=0 i.e. radial flow at inlet.

and the Maximum W.D./unit wt = V . v /g

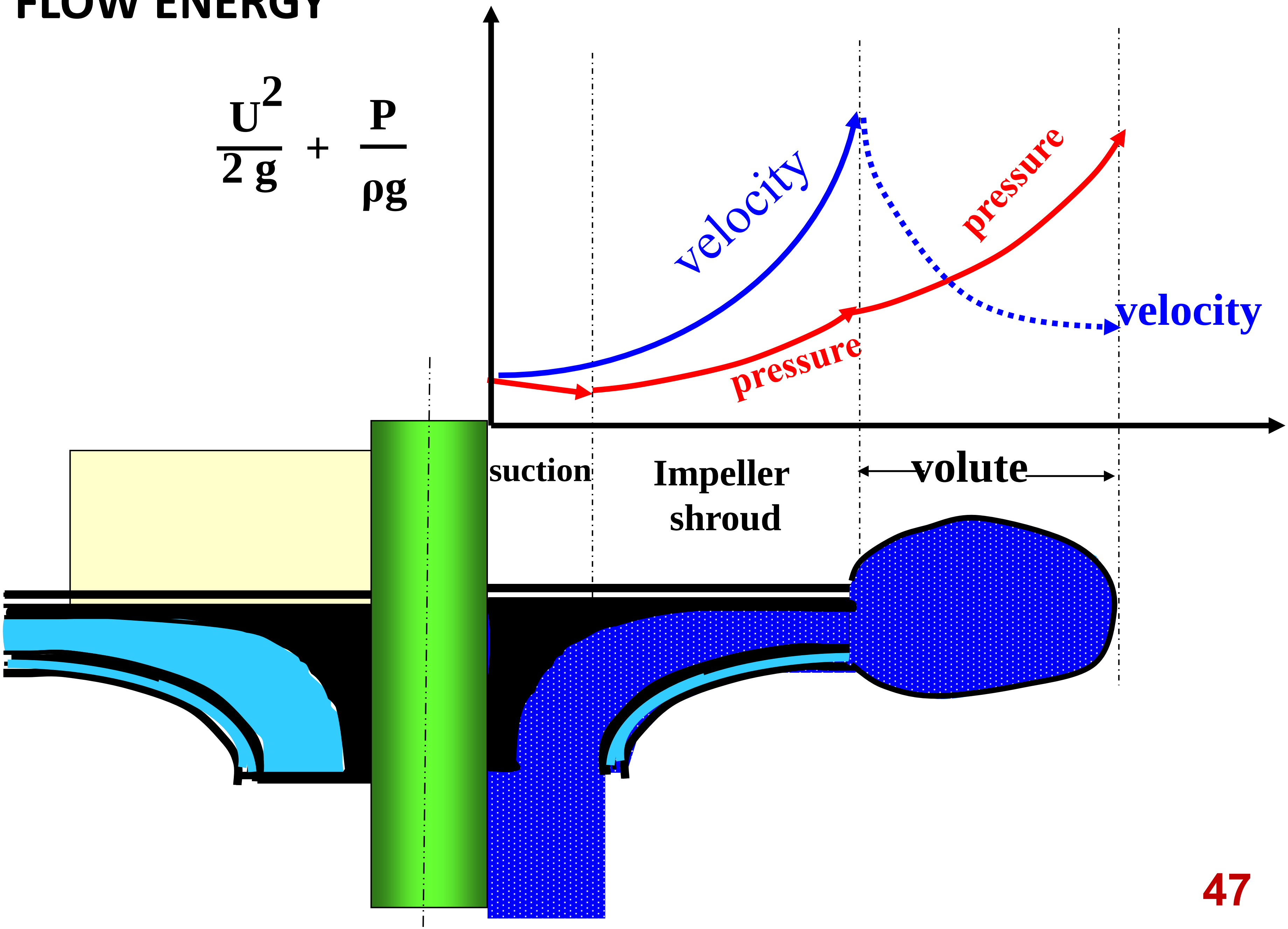
Impeller head

Neglecting Whirl velocity component at inlet ,radial inlet flow, $v_1=0$

- **W.D. on liquid=Eo-Ei;**
- **$Vv/g= U^2/2g+ H_{imp} - U_1^2/2g$ (Neglect $U_1^2/2g$)**
- **$U^2 = v^2+y^2 = (v-y\text{Cot}\,\gamma)^2 + y^2$**
- **$H_{imp}.=(v^2-y^2\,\text{Cosec}^2\,\gamma)/2g$**
- **With $v=\pi DN/60$ & $Y=Q/(\pi DB)$=flow vel.**
- **then $H_{imp}.=C_1N^2-C_2Q^2$**
- **&$H_{casing}=k\,U^2/2g$**
- **$H_{pump}=H_{imp} + H_{casing}$- Hydraulic Loss in both**

FLOW ENERGY

$$\frac{U^2}{2\,g} + \frac{P}{\rho g}$$



<u>OTHER LOSSES</u>:

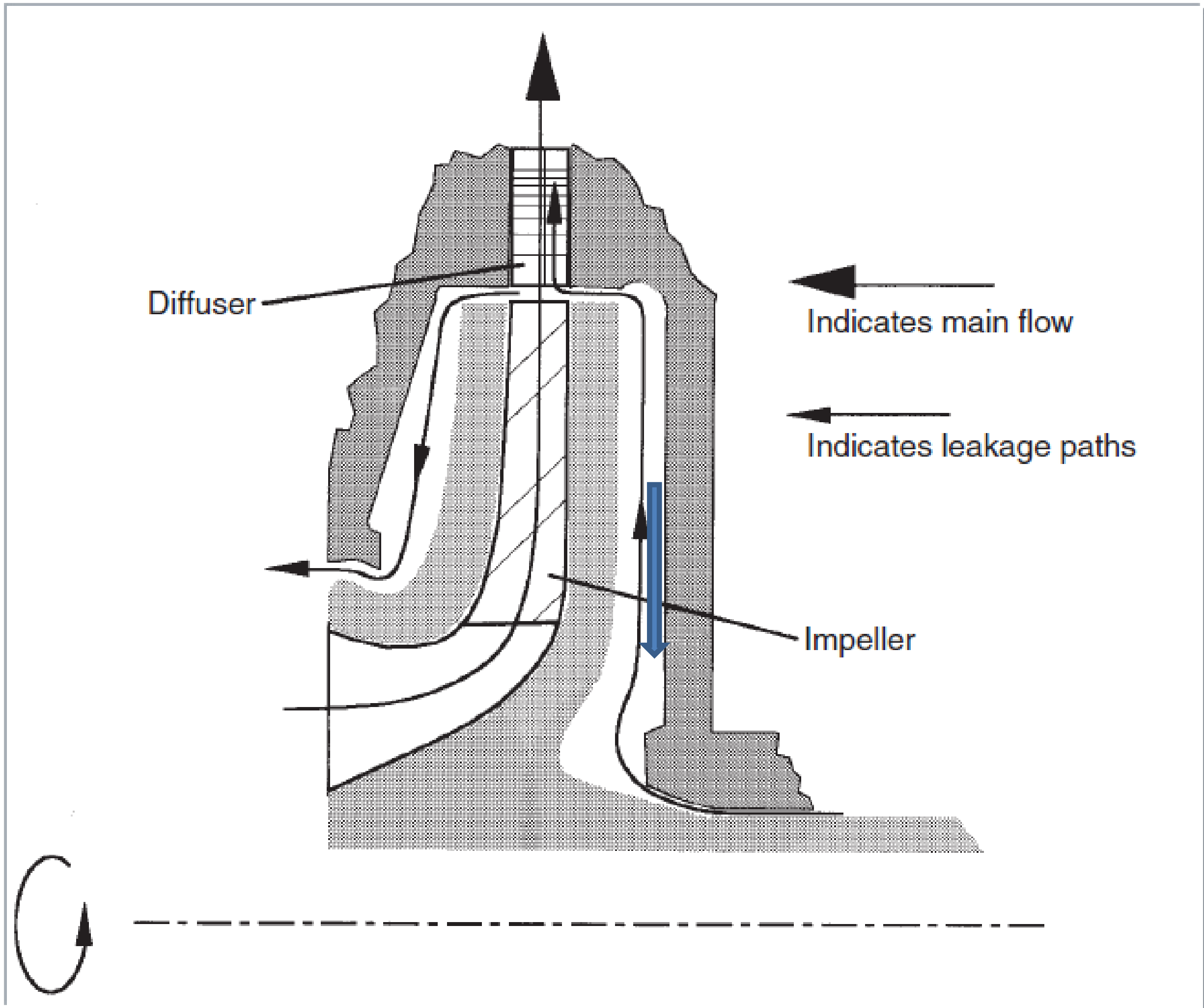
- **Volumetric – leakage through internal and external clearances**
- **Mechanical losses: disk friction, bearings, coupling.**

The sum of ALL losses takes away from the available power delivered by the driver:

$$\boxed{\Sigma_{losses} = \text{Hydraulic} + \text{Volumetric} + \text{Mechanical}}$$

Pump Overall efficiency is:

$$\eta_{overall} = wQH_m / P_{sh}$$



internal leakage in C.P.

The leakage loss, for the purposes of obtaining a numerical estimate, may be regarded as: $Q_L = C_L a_L (2gH_L)^{0.5}$

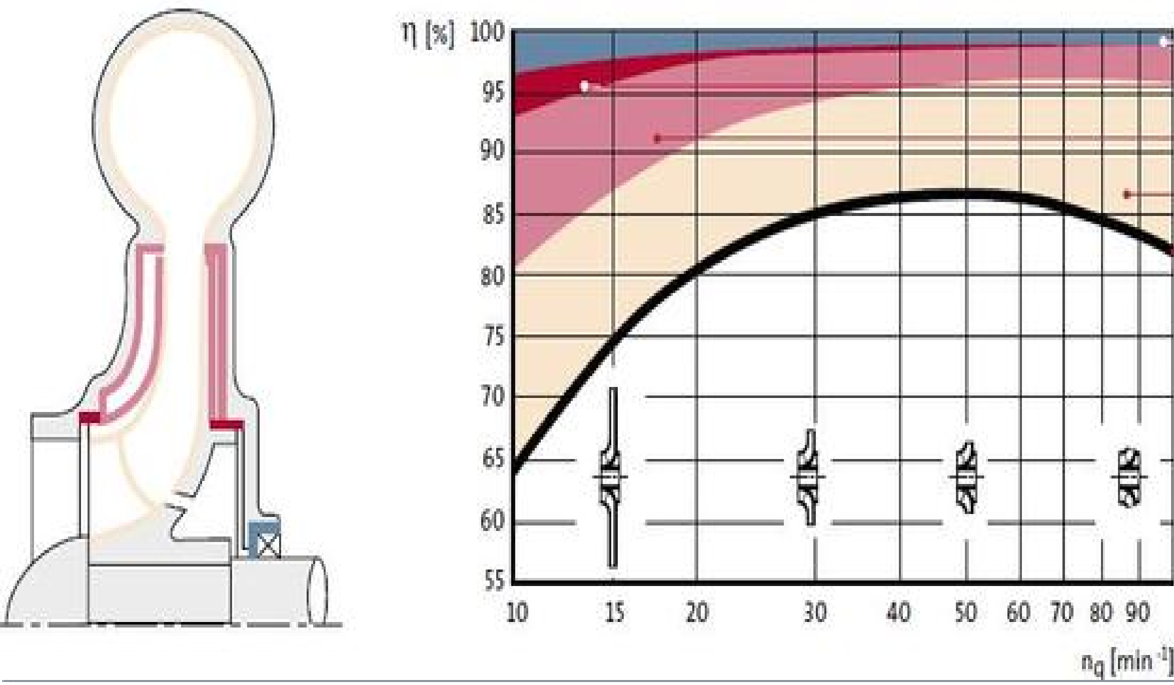


Figure 5.18: Loss distribution in a centrifugal pump as function of specific speed n_q (Ludwig et al., 2002).

ϕ

- Speed ratio ϕ

$$v = \frac{\pi DN}{60} = \phi\sqrt{2gH},$$

Flow ratio: ψ

$$Y = \frac{Q}{\pi DB} = \psi\sqrt{2gH}$$

Specific Speed (S.I. units)

$$N_s = \frac{N\sqrt{P_o}}{H^{1.25}}$$

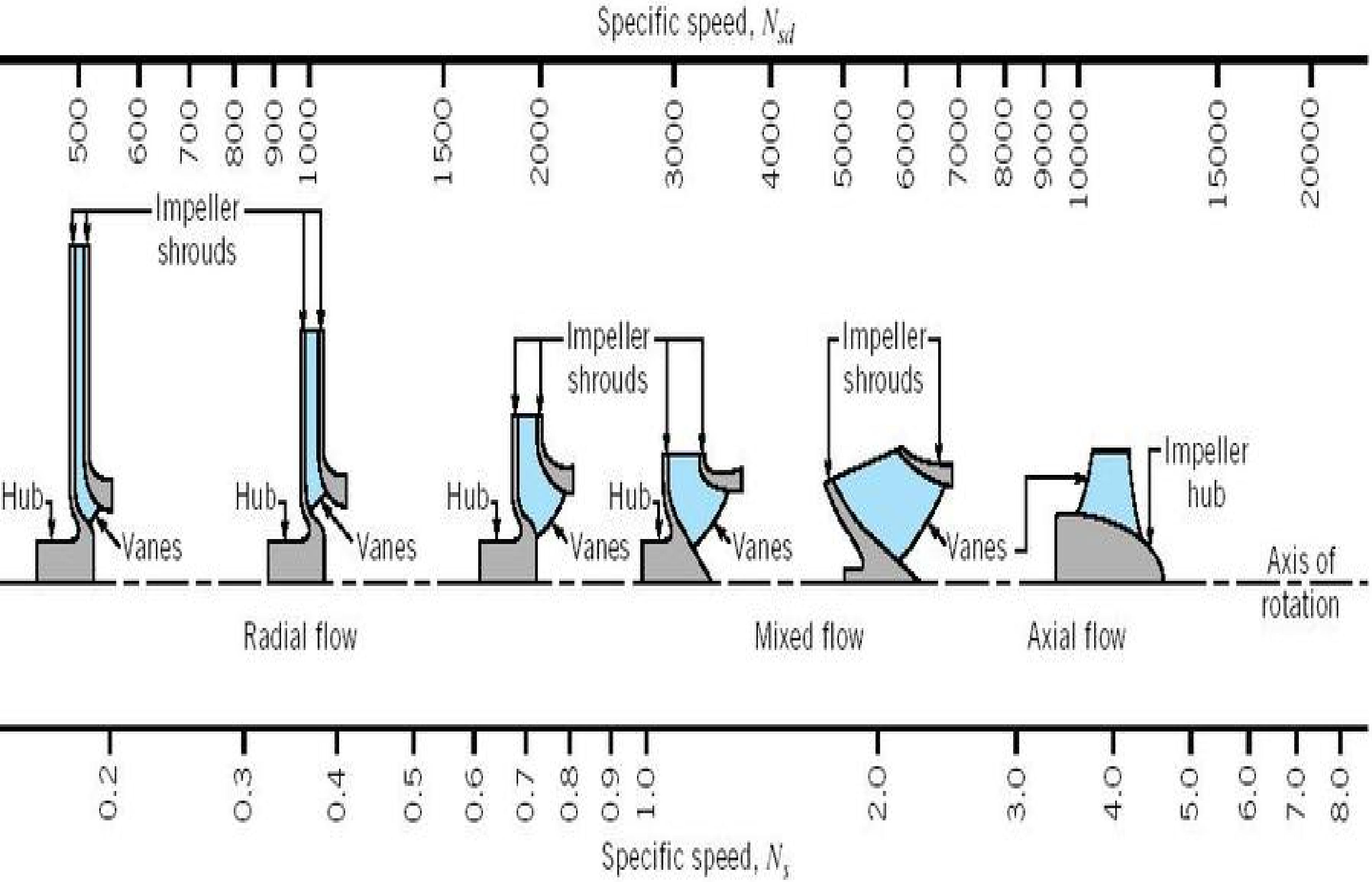
Design Key of Hydraulic Machines

Hydraulic , Manometric Efficiency :

$$\eta_{Hy.,Manometric} = \frac{gH_m}{Vv}$$

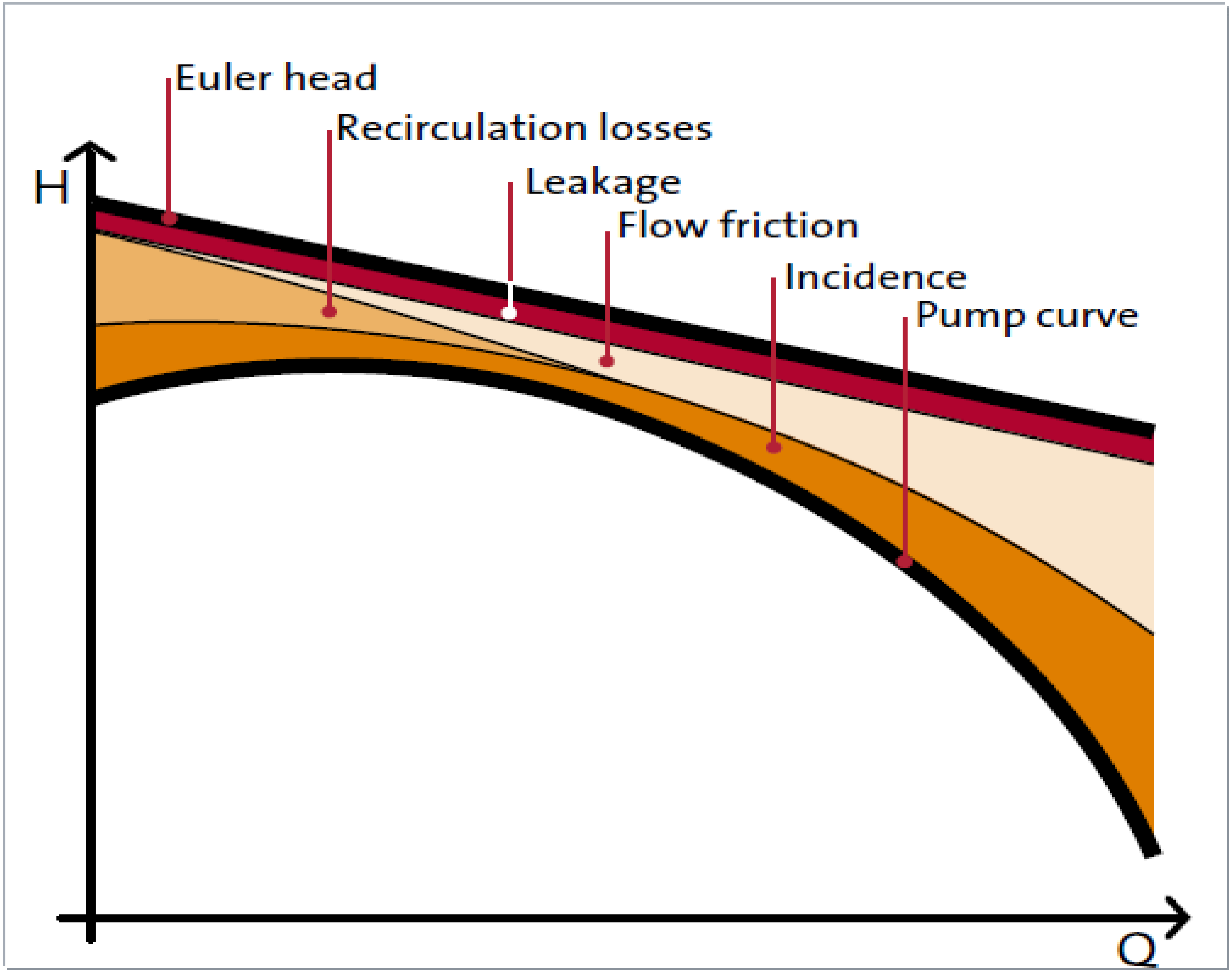
Pump overall Efficiency:

$$\eta_{pump,Overall} = \frac{wQH_m}{P_{sh}}$$



Effect of N_s on Pump Impeller Shape

Theoretical & Real Pump Head Curve

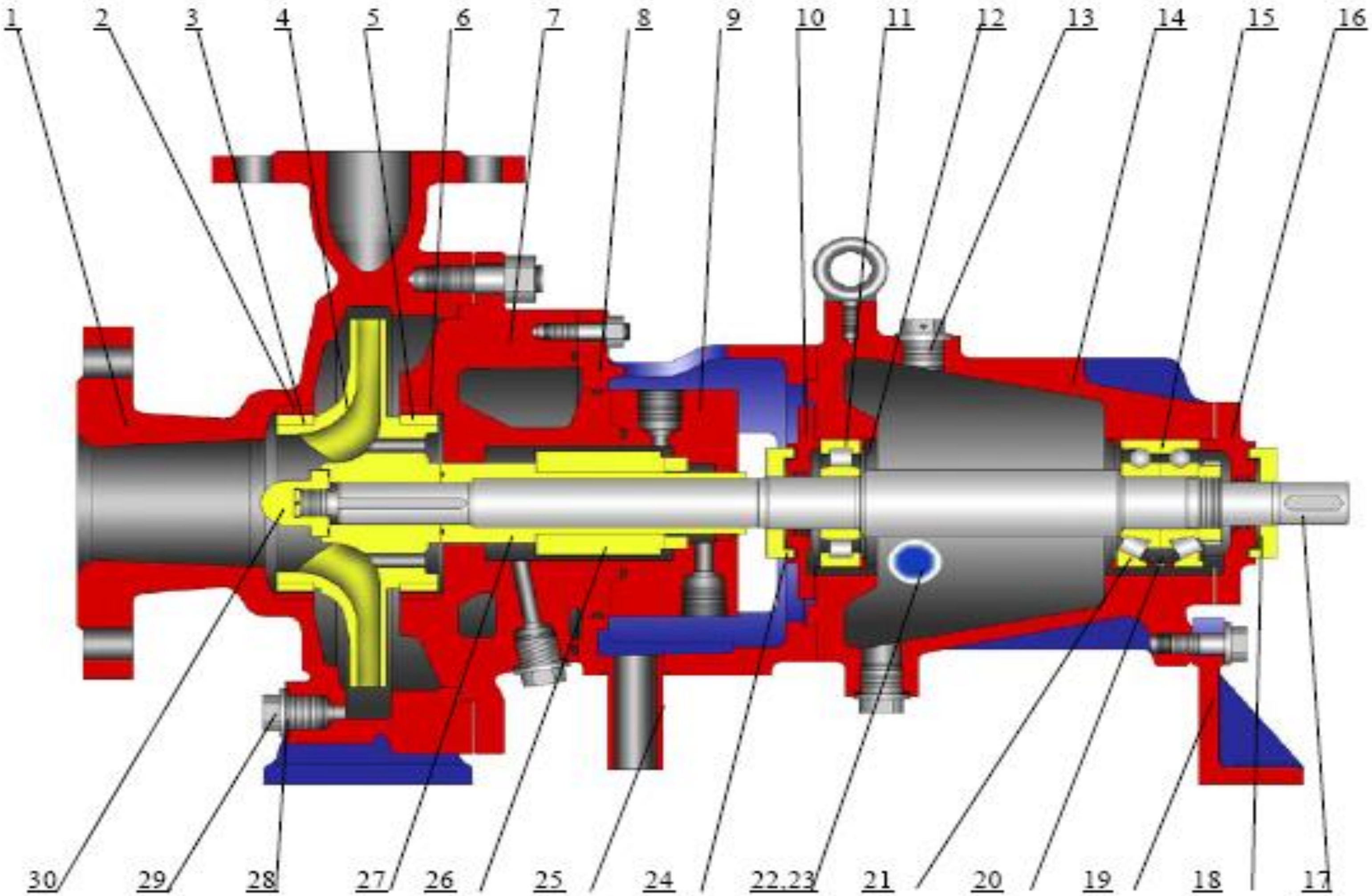


Design Criteria for C.P. impeller

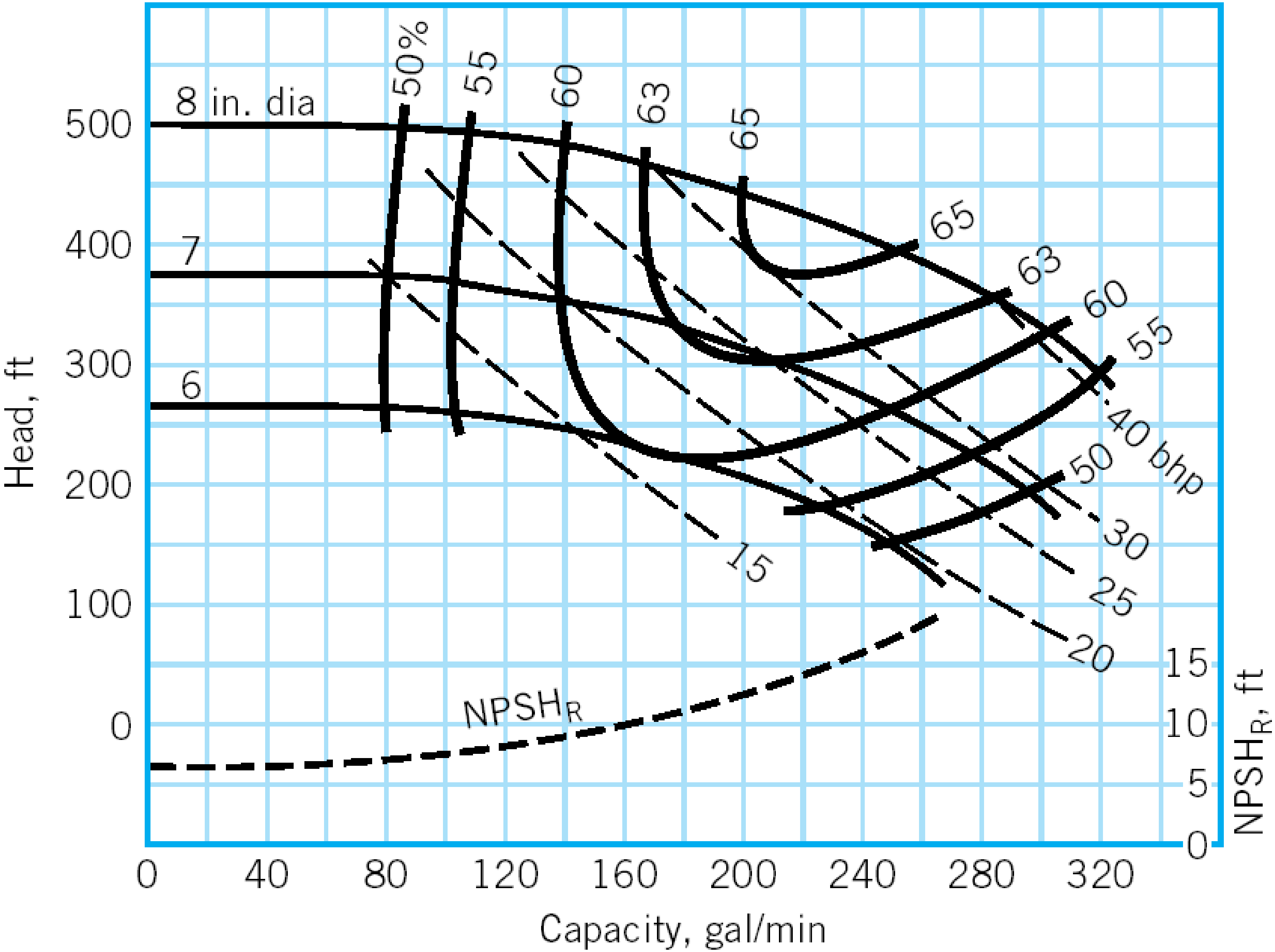
- Outlet blade angle γ
- Inlet blade angle β
- Optimum number of blades:
- $\mathbf{D_i/D_o=}$
- $\mathbf{B_o/D_o=}$
- $\boldsymbol{\eta}_{\mathbf{manometric}}$
- $\boldsymbol{\eta}_{\mathbf{p}}$

4 Cross sectional drawing

Basic design



1Casing	7Casing cover	13 Vent plug	19 Foot	25 Draining pipe
2Casing wearing ring	8Adaptor	14 Bearing housing	20 Bearing spacer ring	26 Mechanical seal
3Front Impeller wearing ring	9Gland cover	15 Angular contact ball bearing	21 Taper roller bearing	27 Shaft sleeve
4Impeller	10Front Bearing cover	16 Back bearing cover	22 Constant level oiler	28 Plug
5Back Impeller wearing ring	11Cylindrical roller bearing	17 Shaft	23 Oil sight glass	29 Sealing gasket
6Casing cover wearing ring	12Snap ring	18 Back deflector	24Front deflector	30 Impeller nut

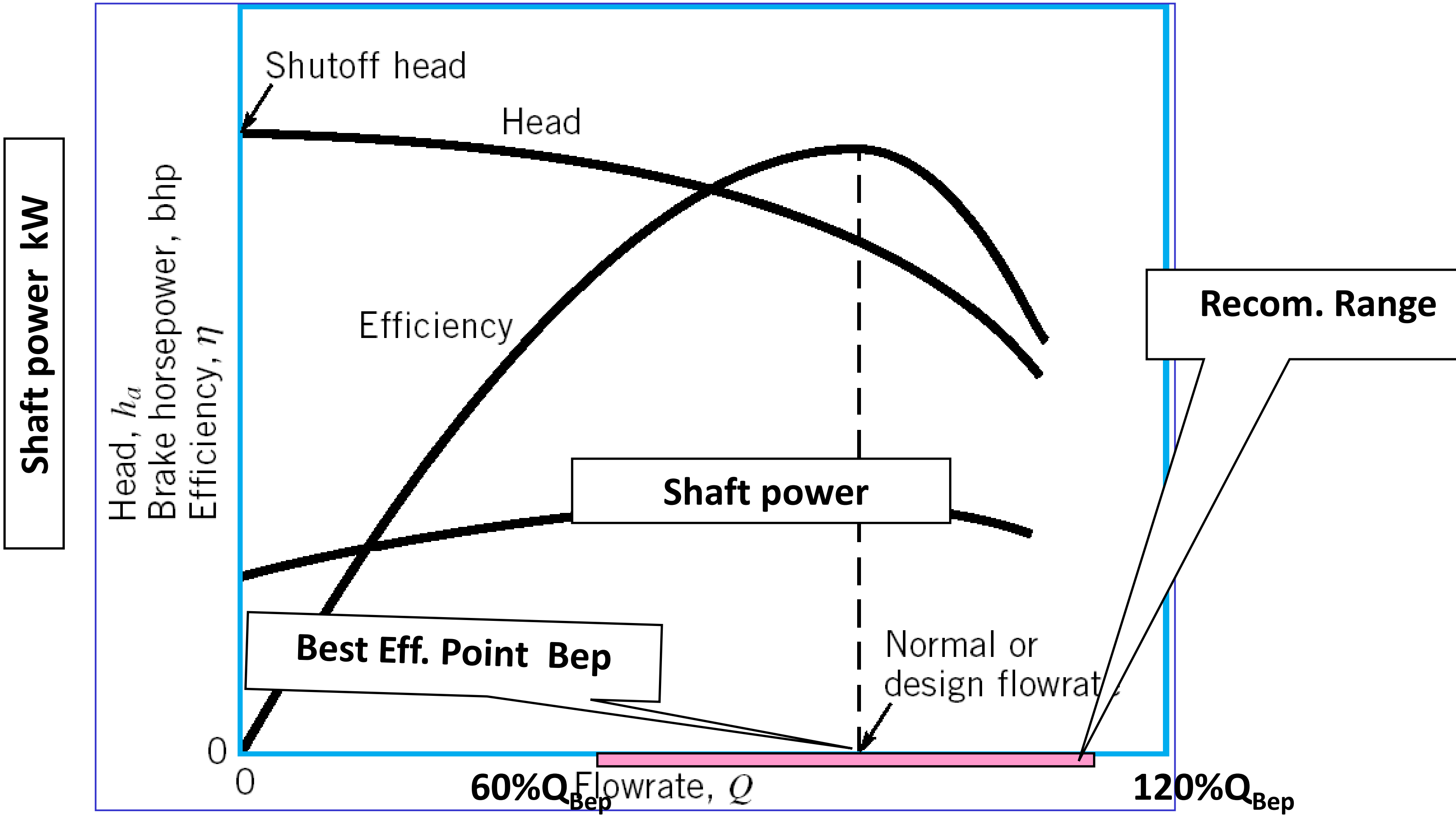


Centrifugal Performance pump Curves

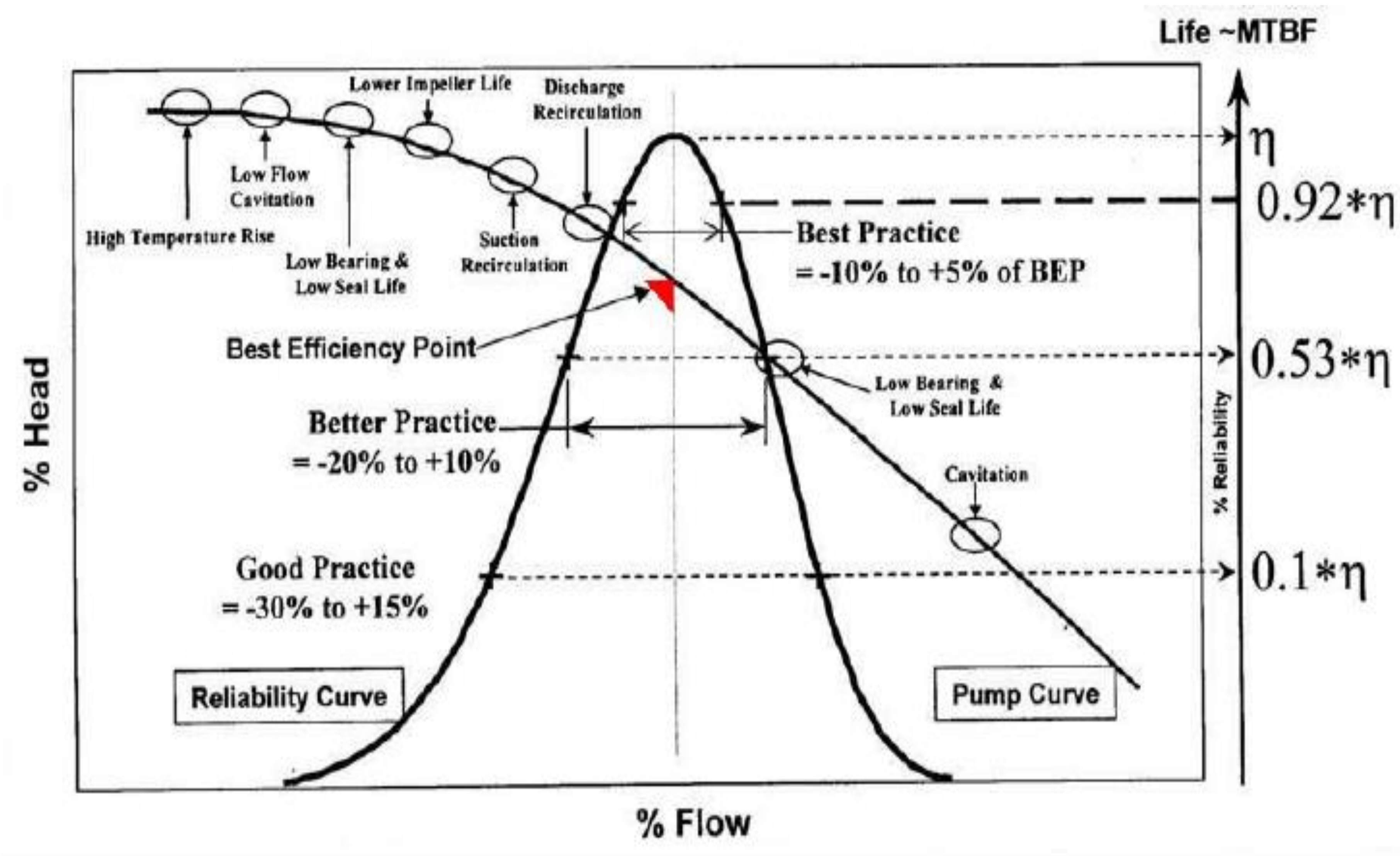
Recommended Operating range: 60-120% of Q Bep

Excessive Noise & vibrations at lower flows

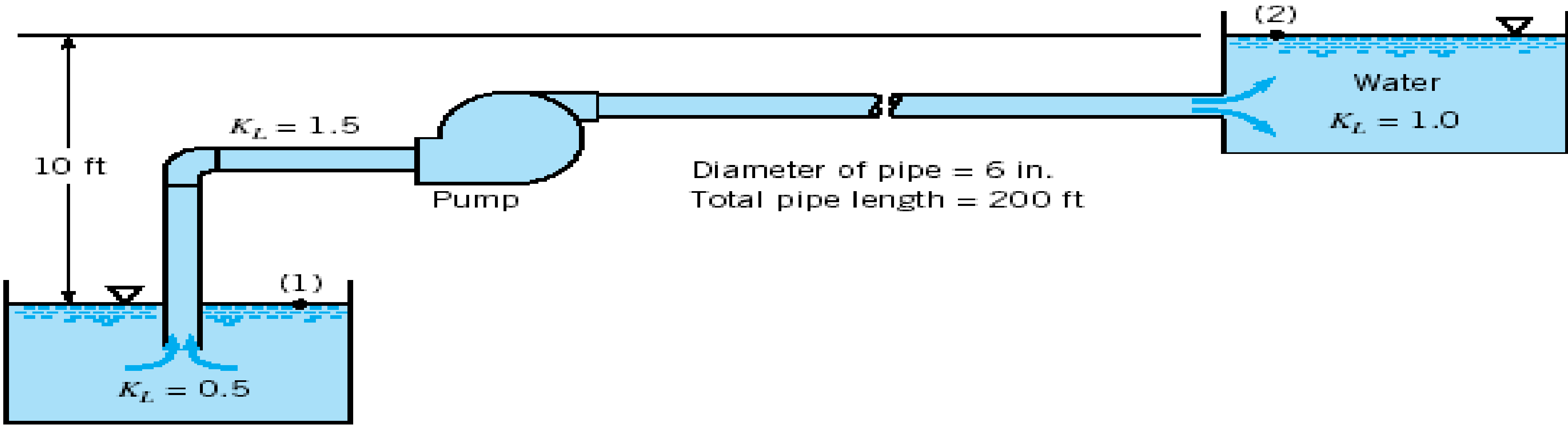
Cavitation expected at higher flows



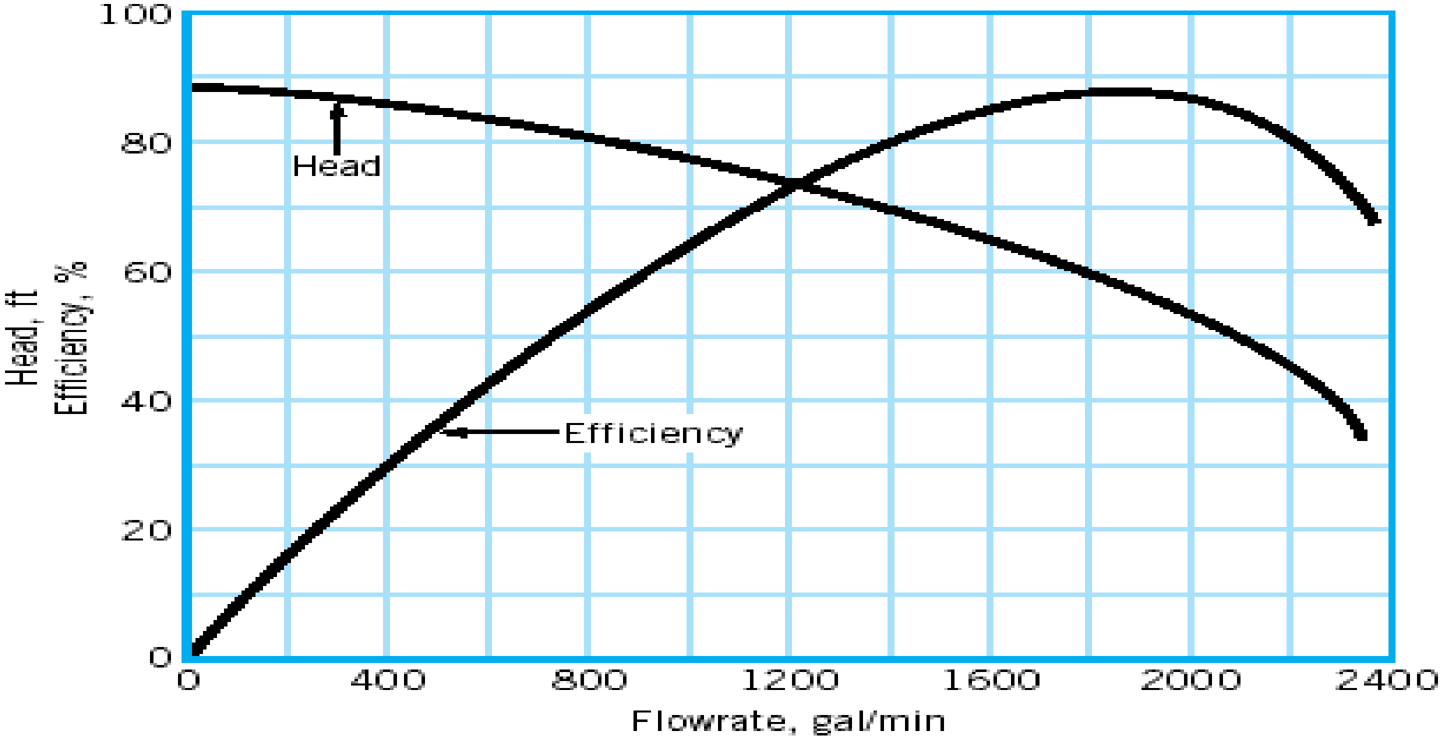
Reliability vs. Relation to Best Efficiency Point



Centrifugal Pump operating Point in Certain Piping System


(2)
Water
$K_L = 1.0$
$K_L = 1.5$
10 ft
Pump
Diameter of pipe = 6 in.
Total pipe length = 200 ft
(1)
$K_L = 0.5$

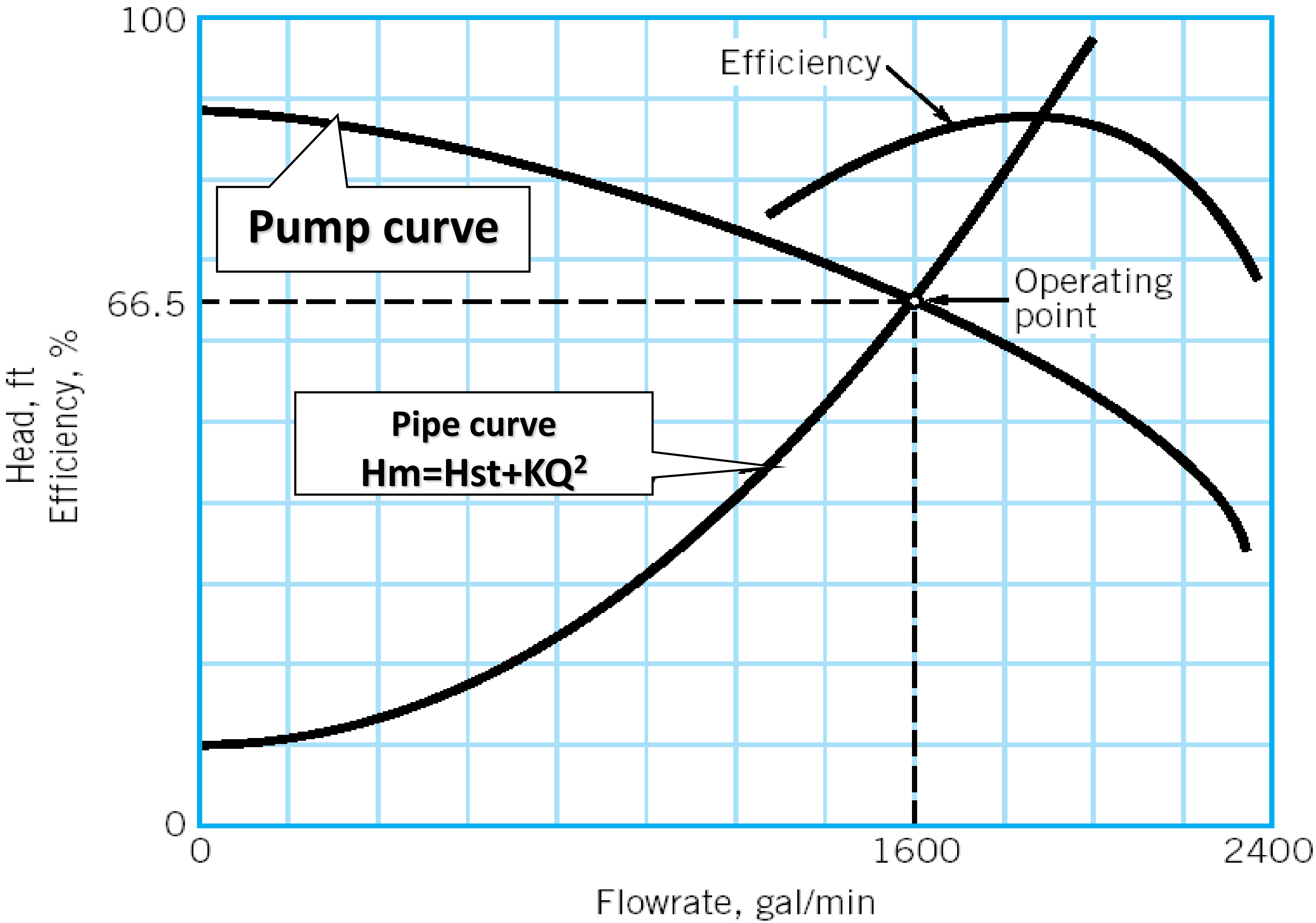
(a)

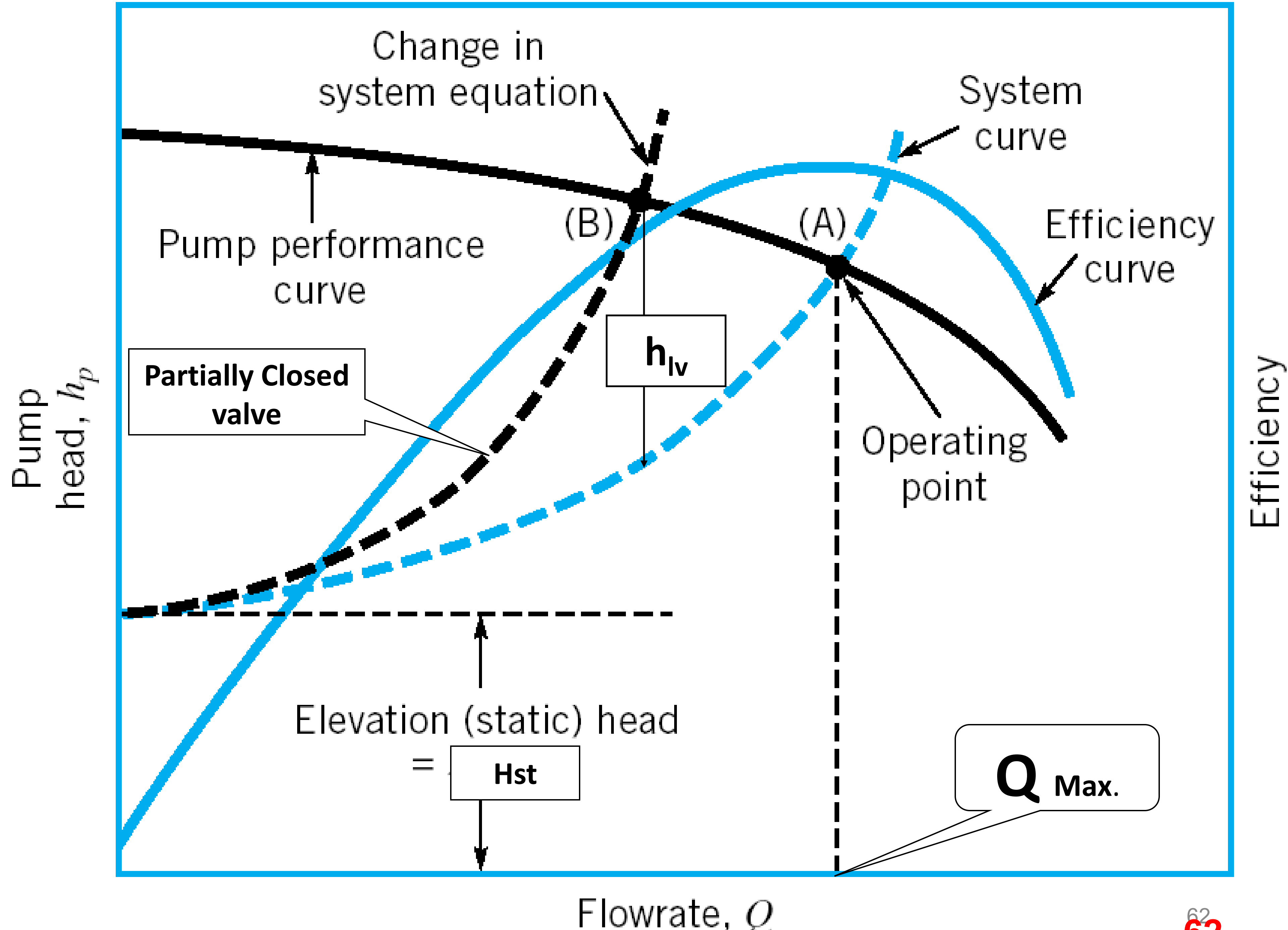

100
80
60
40
20
0
Head, ft
Efficiency, %
Head
Efficiency
0
400
800
1200
1600
2000
2400
Flowrate, gal/min

(b)

Operating Point of C.P. in certain Pipeline

Piping System Head : $H_m = H_{st} + KQ^2$




Change in
system equation
System
curve
Pump performance
curve
(B)
(A)
Efficiency
curve
h_{lv}
Partially Closed
valve
Operating
point
Pump
head, h_p
Efficiency
Elevation (static) head
= Hst
Q Max.
Flowrate, Q

- **From Diagram at operating point (Qmax.)**
- **P_{sh} =For delivery partially opened**
- $H_{lv} = H_{pump} - H_{pipe}$
- $P_{sh\ lv} = wQH_{lv} / \eta_{pump}$
- **Energy waste in valve =**
- **$P_{sh\ lv}$. Working hrs/year ; kWhr.**
- **Delivery valve wastes energy when used to control flow = excess running cost.**
- **Cost of total Energy consumed in pump =**

  $\rho.g.Q.Hm\ T$ (hrs)* Cost of kWhr/($\eta_{pump} . \eta_{motor}$)

MOTOR PUMPS POWER

6.1.3 Motors shall have power ratings, including the service factor (if any), at least equal to the percentages of power at pump rated conditions given in Table 11. However, the power at rated conditions shall not exceed the motor nameplate rating. If it appears that this procedure will lead to unnecessary oversizing of the motor, an alternative proposal shall be submitted for the purchaser's approval.

Table 11 — Power ratings for motor drives

Motor nameplate rating		Percentage of rated pump power
kW	(hp)	%
< 22	(< 30)	125
22 to 55	(30 to 75)	115
> 55	(> 75)	110

Centrifugal Pump Impellers

Open
For Liquids +Impurities

Semi-Open

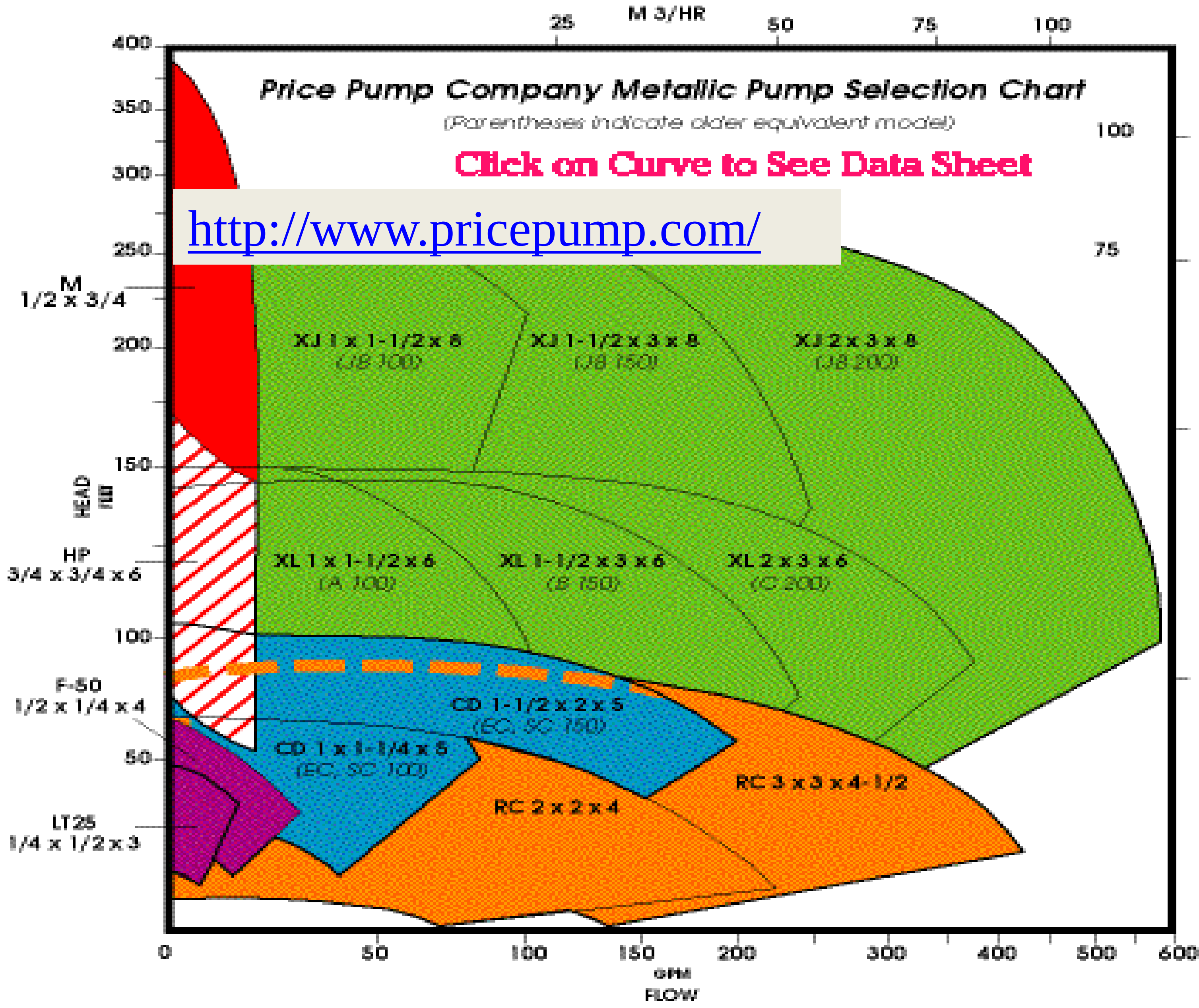
Closed
(for clean Liquids)

Impellers

Pump Selection Chart

Model M

Model X

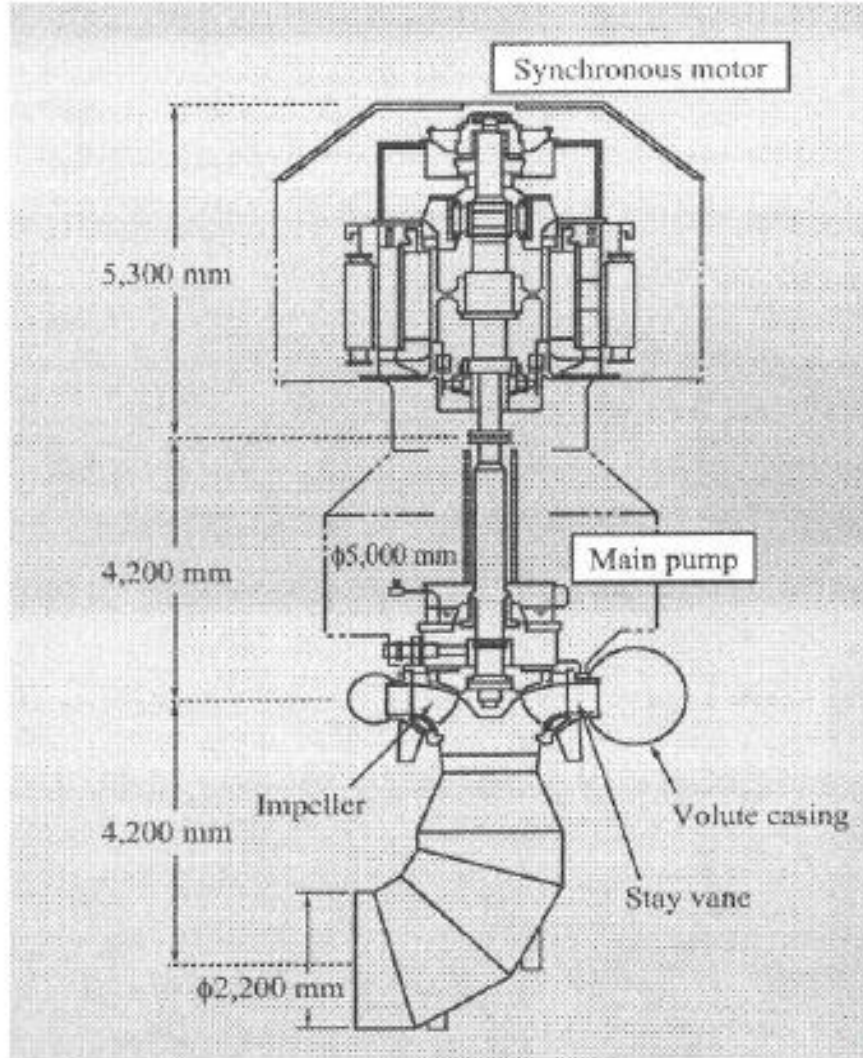


http://www.pricepump.com/

PUMP DESIGN

- MAIN PUMP

Model	Vertical shaft, centrifugal pump
Pump bore	2,400 -1, 800 (mm)
Discharge volume	16.7 m3/s
Speed	210 – 300 min^{-1}
Drive motor	12,000-kW synchronous motor
Number of units	24



- SYNCHRONUS MOTOR

Similarity of Pumps

- **Complete test on large pumps before installation is costly, time consuming and also difficult as the head , flow rate and shaft power cannot be varied easily, especially at the design stage**
- **Tests on a geometrically similar model size under all possible conditions in addition to the Similarity relations obtained by applying Buckingham π theory of dimensional analysis , the Performance of large size pumps and/or at different speeds can be predicted.**

Application of π theory in Pumps to get Similarity relations

- 1-Define the problem and- write the variables

dimensions in terms of M,L,T as;

$D=L$, $N=1/T$, $Q=L^3/T$, $gH=L^2/T^2$,
$\rho=M/L^3$, $\mu=M/LT$, $\varepsilon=L$

2- Collect these variables in π groups using Buckingham theory of dimensional analysis as;

$\pi_1 = gH/N^2 D^2$,$\pi_2 = Q/ND^3$,,
$\pi_3 = \rho ND^2/\mu$, $\pi_4 = \varepsilon/D$ or
, gH/N^2D^2 =Function of $(Q/ND^3, \rho ND^2/\mu, \varepsilon/D)$

Experience **indicated that:**

gH/N^2D^2 =Function of (Q/ND^3)

For geometrically similar pumps under dynamic similar conditions;

- (Q/ND^3)model = (Q/ND^3) pump ,
- (gH/N^2D^2)model = (gH/N^2D^2) pump
- **Scale Effect** ε/D)model> ε/D) pump, then $\eta_p > \eta_m$ **use empirical formula to get** η_p **as;**
- **Moody formula**

$$\frac{1-\eta p}{1-\eta m} = \left(\frac{Dm}{Dp}\right)^{0.2}$$

- **Ackerat formula**

$$\frac{1-\eta p}{1-\eta m} = \frac{1}{2}\left\{1+\left(\frac{Dm}{Dp}\right)^{0.2}\left(\frac{Hm}{Hp}\right)^{0.1}\right\}$$

Affinity Law
Pump Speed Variation

- For the same pump under dynamically similar conditions ,substitute D=Constant, in the previous formulas, (constant efficiency)
-
- $$\frac{N_2}{N_1} = \frac{Q_2}{Q_1} = \sqrt{\frac{H_2}{H_1}} = \sqrt[3]{\frac{P_2}{P_1}}$$ **with $\eta_1 = \eta_2$**
- These relations can be used to obtain the performance curves of C.P. at any speed when they are known at certain speed.

Centrifugal Pump Characteristics

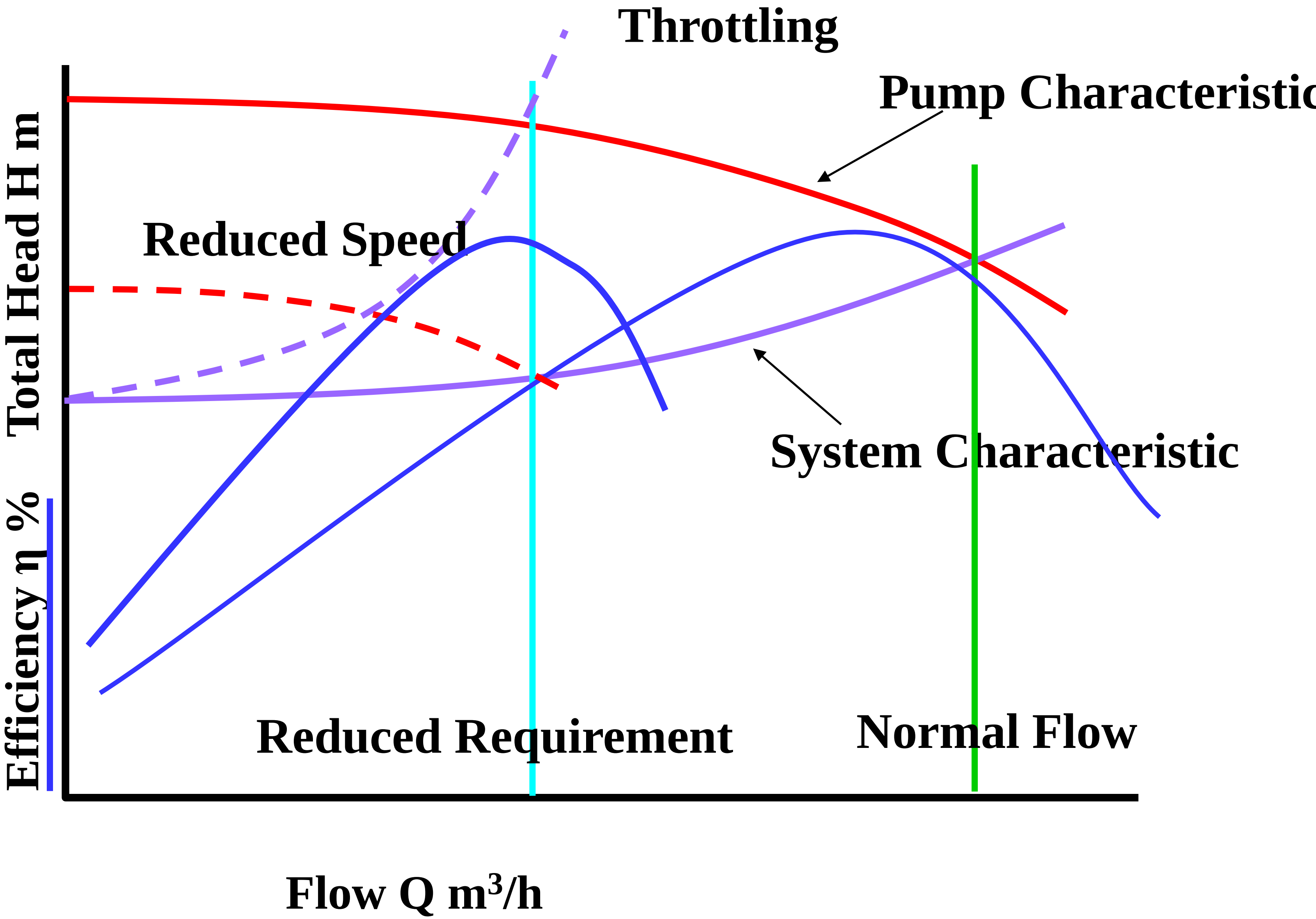


Table 10.1 Power Requirements for Constant- and Variable-Speed Drive Pumps

Throttle Valve Control with Constant-Speed (1750 rpm) Motor

Flow Rate (gpm)	System Head (ft)	Valve[a] Efficiency (%)	Pump Head (ft)	Pump Efficiency (%)	Pump Power (bhp)	Motor Efficiency (%)	Motor Input (hp)	Power Input[b] (hp)
1700	180	100.0	180	80.0	96.7	90.8	106.5	106.7
1500	150	78.1	192	78.4	92.9	90.7	102.4	102.6
1360	131	66.2	198	76.8	88.6	90.7	97.7	97.9
1100	102	49.5	206	72.4	79.1	90.6	87.3	87.5
900	83	39.5	210	67.0	71.3	90.3	79.0	79.1
600	62	29.0	214	54.0	60.1	90.0	66.8	66.9

Variable-Speed Drive with Energy-Efficient Motor

Flow Rate (gpm)	Pump/ System Head (ft)	Pump Efficiency (%)	Pump Power (bhp)	Motor speed (rpm)	Motor Efficiency (%)	Motor Input (hp)	Control Efficiency (%)	Power Input (hp)
1700	180	80.0	96.7	1750	93.7	103.2	97.0	106.4
1500	150	79.6	71.5	1580	94.0	76.0	96.1	79.1
1360	131	78.8	57.2	1470	93.9	60.9	95.0	64.1
1100	102	78.4	36.2	1275	93.8	38.6	94.8	40.7
900	83	77.1	24.5	1140	92.3	26.5	92.8	28.6
600	62	72.0	13.1	960	90.0	14.5	89.1	16.3

Source: Based on Armintor and Conners [18].

[a]Valve efficiency is the ratio of system pressure to pump pressure.

[b]Power input is motor input divided by 0.998 starter efficiency.

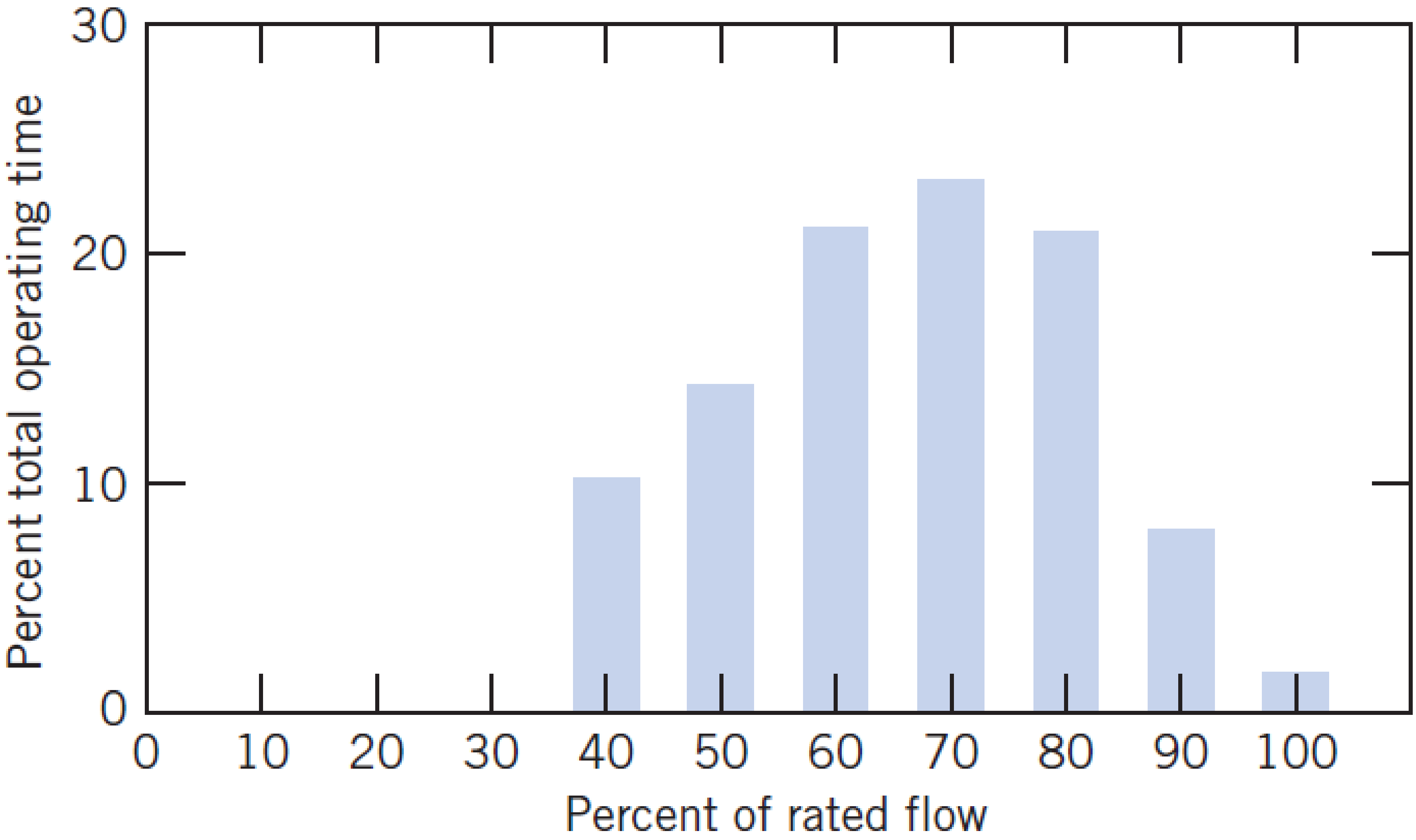
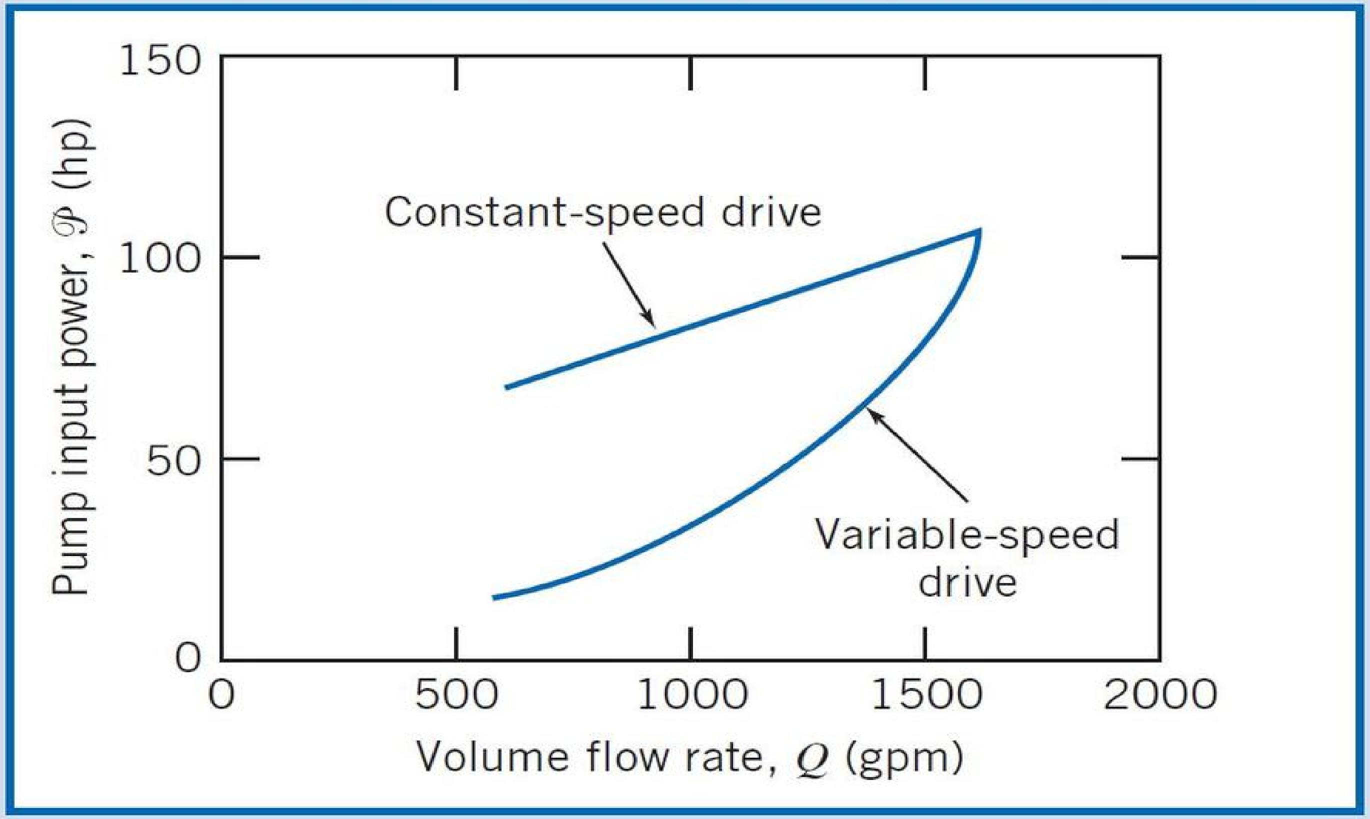


Fig.10.25 : Mean duty cycle for centrifugal pumps in the chemical and petroleum industries [18].

Example : ENERGY SAVINGS WITH VARIABLE-SPEED CENTRIFUGAL PUMP DRIVE

- Combine the information on mean duty cycle for centrifugal pumps given in Fig. 10.25 with the drive data in Table 10.1. Estimate the annual savings in pumping energy and cost that could be achieved by implementing a variable-speed drive system.
- Given: Consider the variable-flow, variable-pressure pumping system of Table 10.1. Assume the system operates on the typical duty cycle shown in Fig. 10.25, 24 hours per day, year round.
- Find: (a) An estimate of the reduction in annual energy usage obtained with the variable-speed drive.
- (b) The energy costs and the cost saving due to variable-speed operation.
- **Solution:** Full-time operation involves 365 days X 24 hours per day, or 8760 hours per year. Thus the percentages in Fig. 10.27 may be multiplied by 8760 to give annual hours of operation.
- First plot the pump input power versus flow rate using data from Table 10.1 to allow interpolation, as shown below



Illustrate the procedure using operation at 70 percent flow rate as a sample calculation. At 70 percent flow rate, the pump delivery is 0.7×1700 gpm = 1190 gpm. From the plot, the pump input power requirement at this flow rate is 90 hp for the constant-speed drive. At this flow rate, the pump operates 23 percent of the time, or $0.23 \times 8760 = 2015$ hours per year. The total energy consumed at this duty point is $90 \text{ hp} \times 2015 \text{ hr} = 1.81 \times 10^5$ hp·hr. The electrical energy consumed is

$$E = 1.81 \times 10^5 \text{ hp}\cdot\text{hr} \times 0.746 \frac{\text{kW}\cdot\text{hr}}{\text{hp}\cdot\text{hr}} = 1.35 \times 10^5 \text{ kW}\cdot\text{hr}$$

The corresponding cost of electricity [at \$0.12/(kW·hr)] is

$$C = 1.35 \times 10^5 \text{ kW}\cdot\text{hr} \times \frac{\$0.12}{\text{kW}\cdot\text{hr}} = \$16{,}250$$

The following tables were prepared using similar calculations:

Constant-Speed Drive, 8760 hr/yr					
Flow (%)	**Flow (gpm)**	**Time (%)**	**Time (hr)**	**Power (hp)**	**Energy (hp·hr)**
100	1700	2	175	109	1.91×10^4
90	1530	8	701	103	7.20×10^4
80	1360	21	1840	96	17.7×10^4
70	1190	23	2015	90	18.1×10^4
60	1020	21	1840	84	15.4×10^4
50	850	15	1314	77	10.2×10^4
40	680	10	876	71	6.21×10^4
				Total:	76.7×10^4

Summing the last column of the table shows that for the constant-speed drive system the annual energy consumption is 7.67×10^5 hp·hr. The electrical energy consumption is

$$E = 7.67 \times 10^5 \text{ hp·hr} \times 0.746 \frac{\text{kW·hr}}{\text{hp·hr}} = 572{,}000 \text{ kW·hr} \leftarrow E_{\text{CSD}}$$

At \$0.12 per kilowatt hour, the energy cost for the constant-speed drive system is

$$C = 572{,}000 \text{ kW·hr} \times \frac{\$0.12}{\text{kW·hr}} = \$68{,}700 \leftarrow C_{\text{CSD}}$$

Variable-Speed Drive, 8760 hr/yr					
Flow (%)	**Flow (gpm)**	**Time (%)**	**Time (hr)**	**Power (hp)**	**Energy (hp·hr)**
100	1700	2	175	109	1.90×10^4
90	1530	8	701	81	5.71×10^4
80	1360	21	1840	61	11.2×10^4
70	1190	23	2015	46	9.20×10^4
60	1020	21	1840	34	6.29×10^4
50	850	15	1314	26	3.37×10^4
40	680	10	876	19	1.68×10^4
				Total:	39.4×10^4

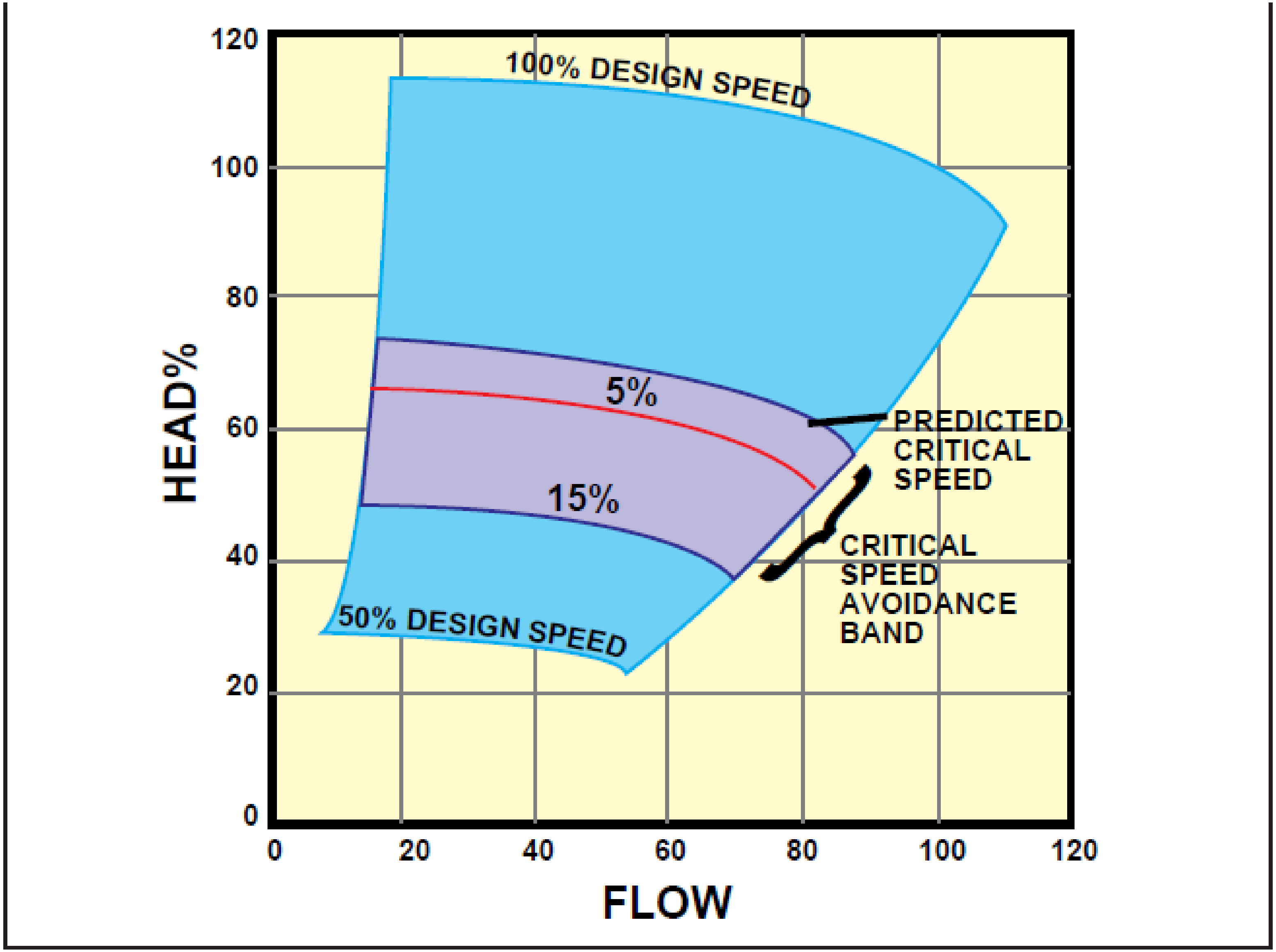
Summing the last column of the table shows that for the variable-speed drive system, the annual energy consumption is 3.94X10^5 hp.hr. The electrical energy consumption is

$$E = 3.94 \times 10^5 \text{ hp}\cdot\text{hr} \times 0.746 \frac{\text{kW}\cdot\text{hr}}{\text{hp}\cdot\text{hr}} = \underbrace{294{,}000 \text{ kW}\cdot\text{hr}}_{E_{\text{VSD}}}$$

At $0.12 per kilowatt hour, the energy cost for the variable-speed drive system is only

$$C = 294{,}000 \text{ kW}\cdot\text{hr} \times \frac{\$0.12}{\text{kW}\cdot\text{hr}} = \underbrace{\$35{,}250}_{C_{\text{VSD}}}$$

Thus, in this application, the variable-speed drive **reduces energy consumption by 278,000 kWhr** (47 percent). The **cost saving is an impressive $33,450 annually**. One could afford to install a variable speed drive even at considerable cost penalty. The savings in energy cost are appreciable each year and continue throughout the life of the system.



Avoiding lateral critical speeds **Why ?**

ASHRAE JOURNAL

Energy Savings From Pump Impeller Trimming

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Member ASHRAE

and

John W. Mitchell, Ph.D., P.E.
Fellow ASHRAE

this study, three building pumps were evaluated for energy savings from impeller trimming: heating pumps, primary chilled water pumps and condenser pumps.

Pump Oversizing Practices

Pump oversizing often stems from prudent engineering practices such as:

Effect of changing the impeller diameter (for the same casing)

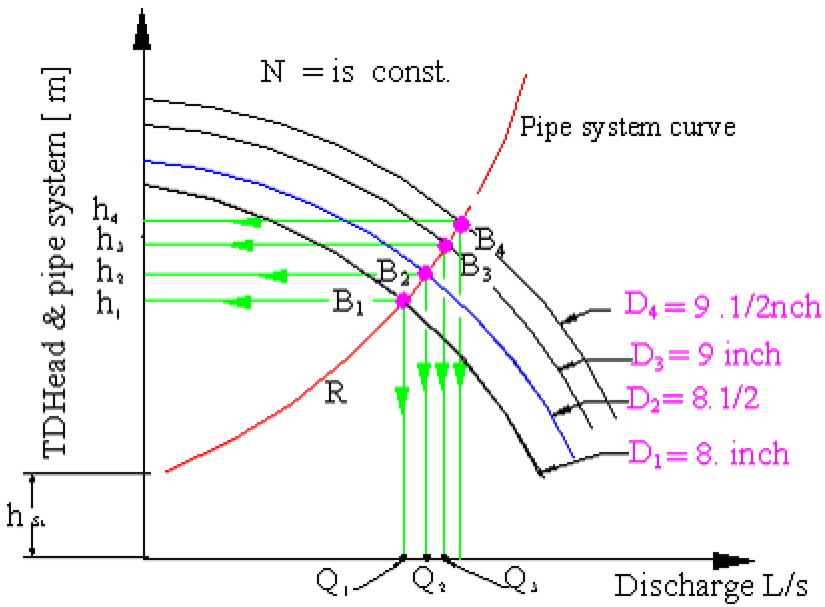
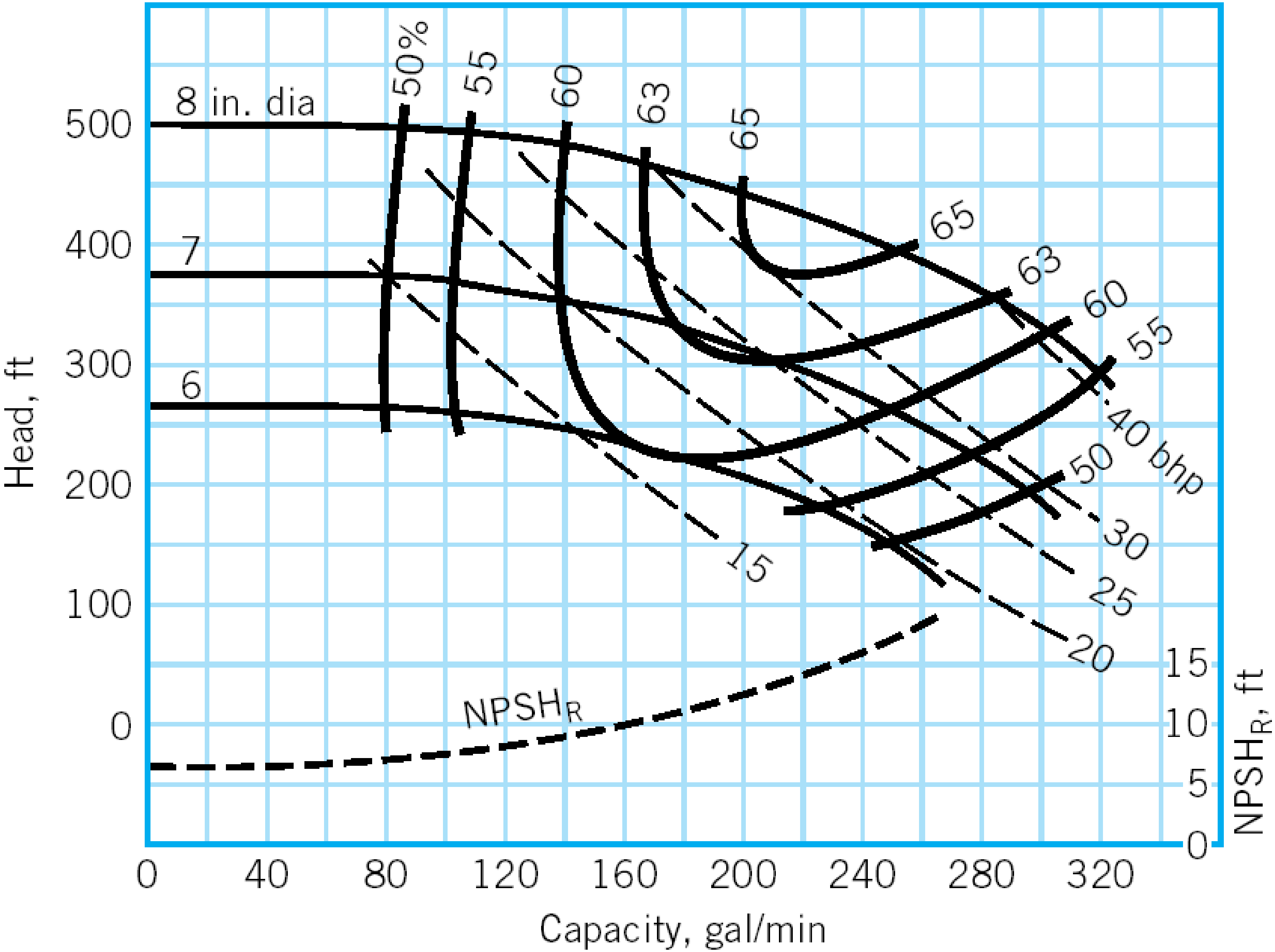
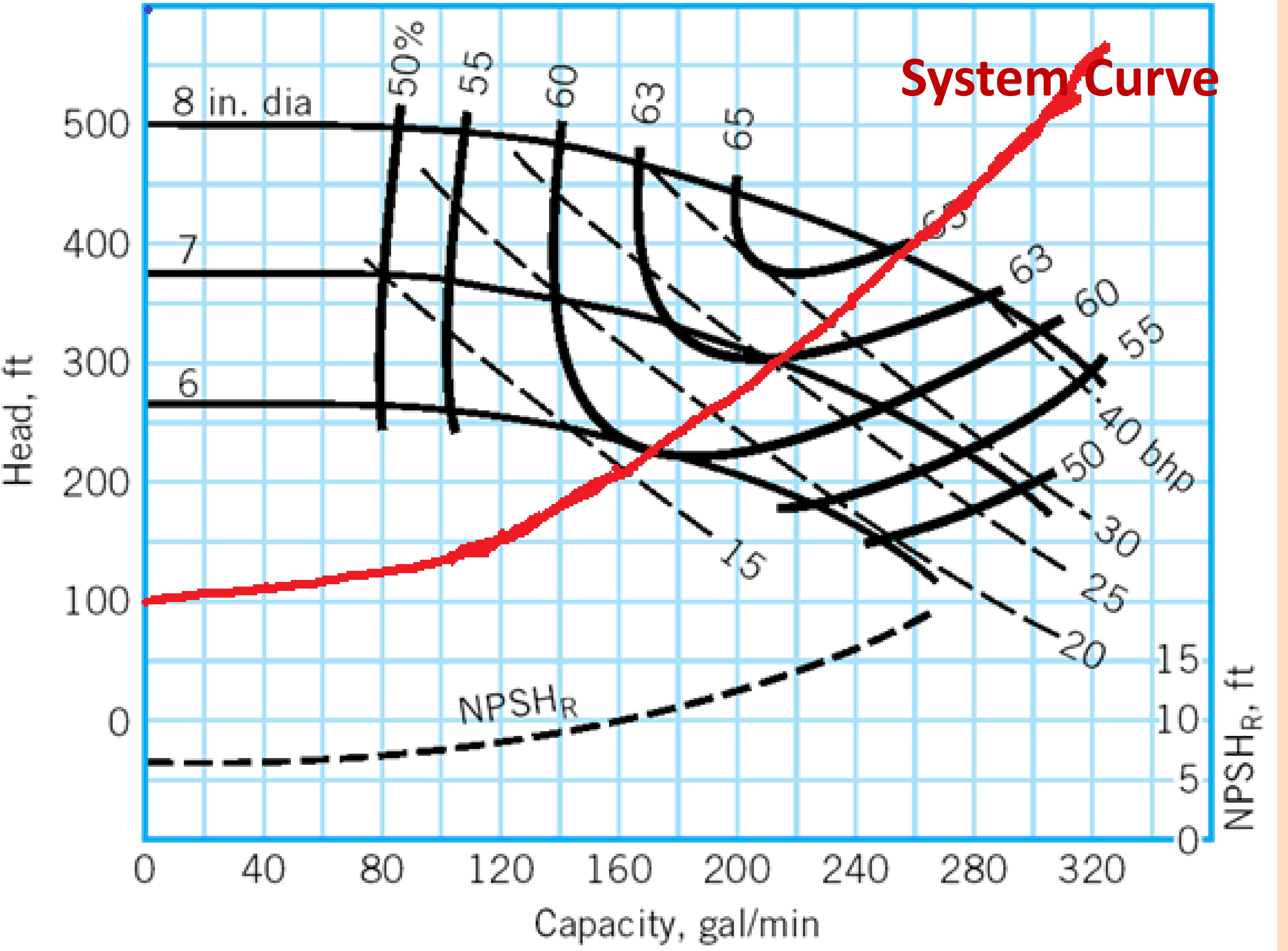


Figure (5.4) Effect of Changing the Impeller diameter



Effect of Impeller Diameter on Centrifugal Performance pump Curves



System Curve
8 in. dia
7
6
50%
55
60
63
65
65
63
60
55
40 bhp
50
30
25
20
15
NPSH$_R$
500
400
300
200
100
0
Head, ft
15
10
5
0
NPSH$_R$, ft
0
40
80
120
160
200
240
280
320
Capacity, gal/min

Trimming Relations: PUMP Hand-Book

1- Capacity varies directly with the impeller diameter:

$$\boxed{\frac{Q_1}{Q_2} = \frac{D_1}{D_2}}$$

2- The total head varies with the square of the impeller diameter:

$$\boxed{\frac{h_{a1}}{h_{a2}} = \left(\frac{D_1}{D_2}\right)^2}$$

3- The power required by the pump varies with the cube of the impeller diameter:

$$\boxed{\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3}$$

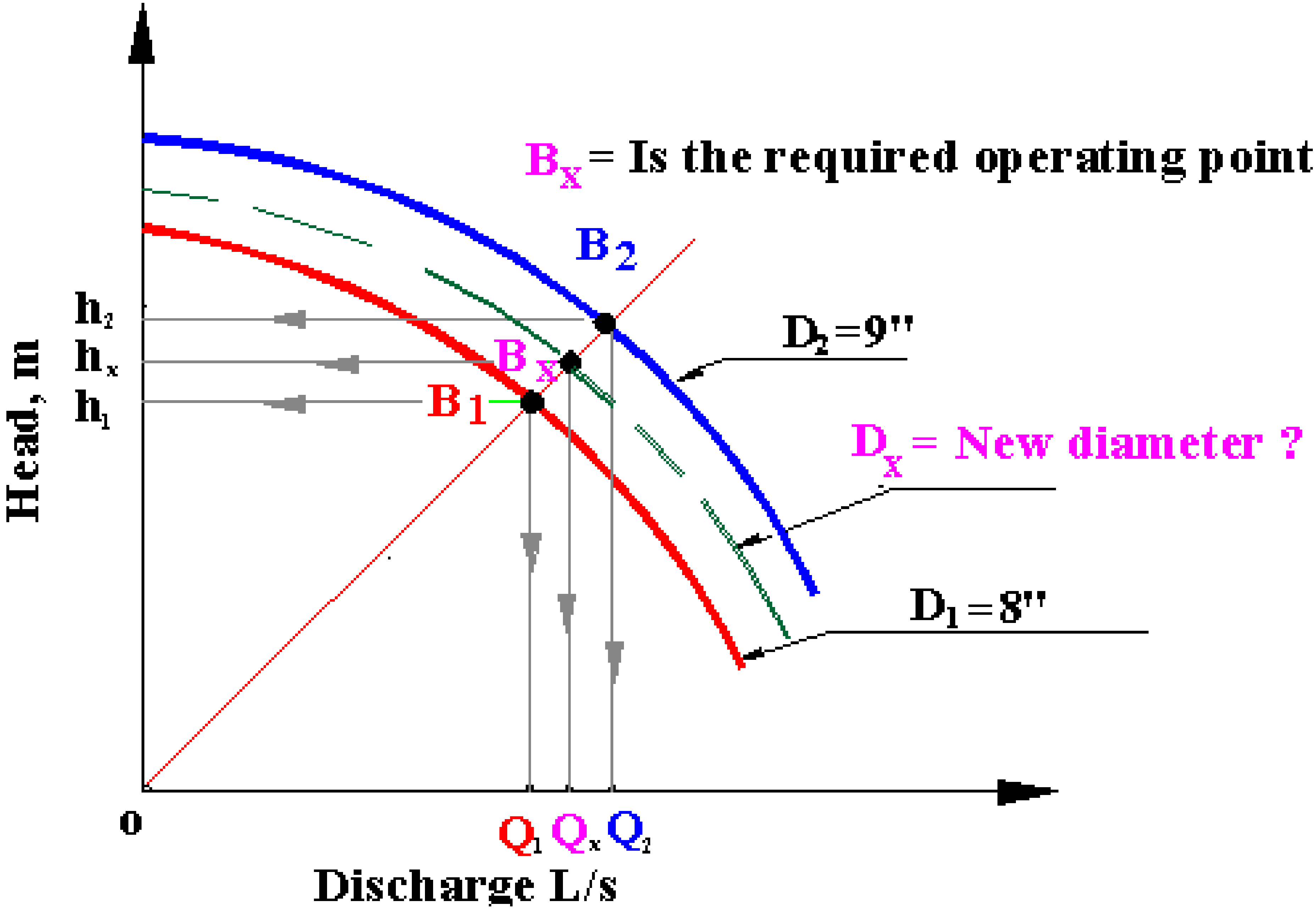
The pump efficiency is expected to drop slightly due to the increases in the clearance between the impeller tip and diffuser. Refer to pump's catalogue

Trimming Relations:
Sulzer Co. Centrifugal Pump Hand-Book

$$\boxed{\frac{Q'}{Q} \approx \frac{H'}{H} \approx \left(\frac{D'}{D}\right)^m}$$

m =2 for ΔD> 6%
m =3 for ΔD<1%
n =1/m

$$D' = D.\left[\frac{Q'}{Q}\right]^n = D.\left[\frac{H'}{H}\right]^n$$



Bx = Is the required operating point
B2
D2 = 9"
h2
hx
h1
Head, m
Bx
B1
Dx = New diameter ?
D1 = 8"
0
Q1
Qx
Q2
Discharge L/s

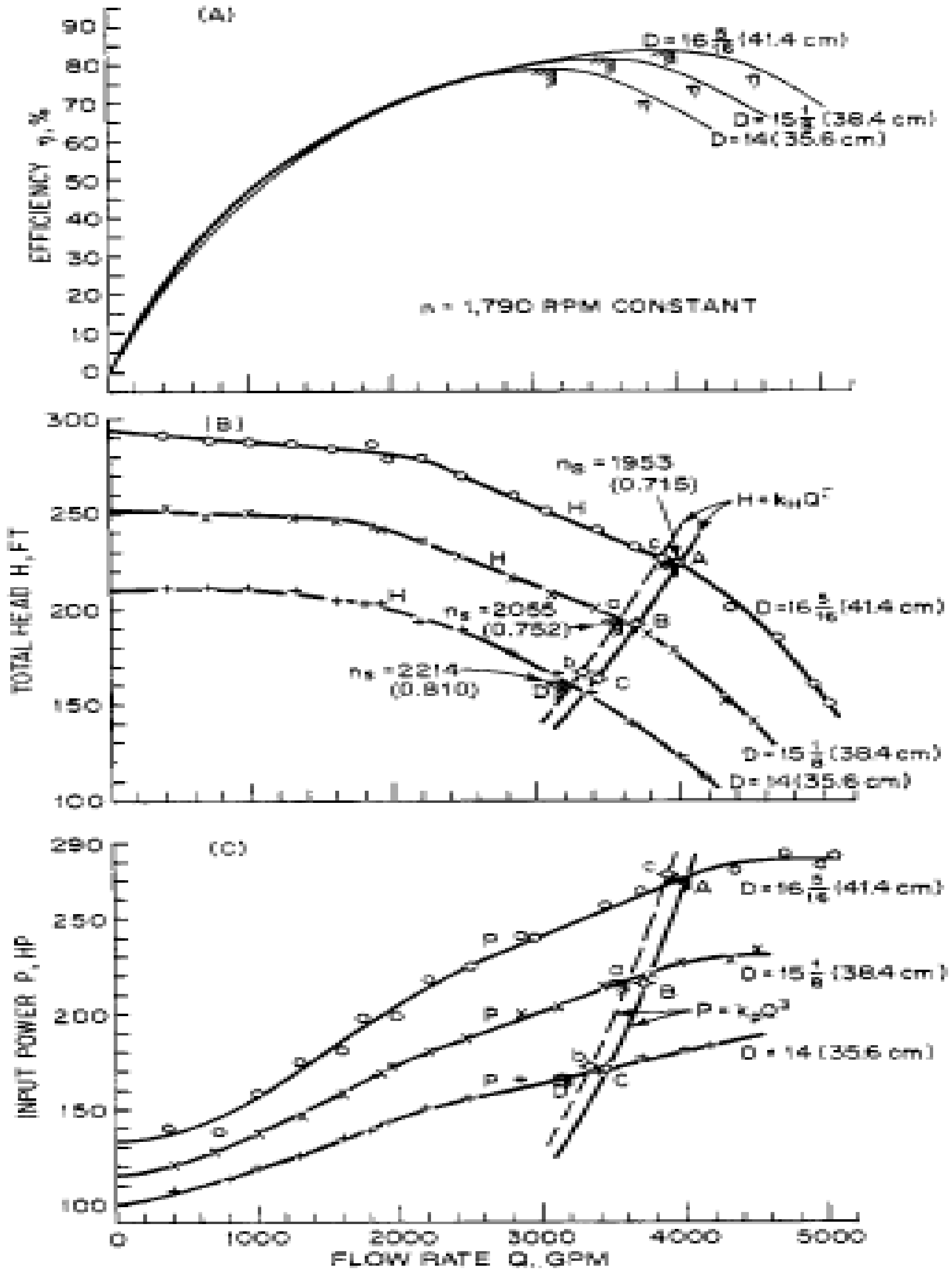
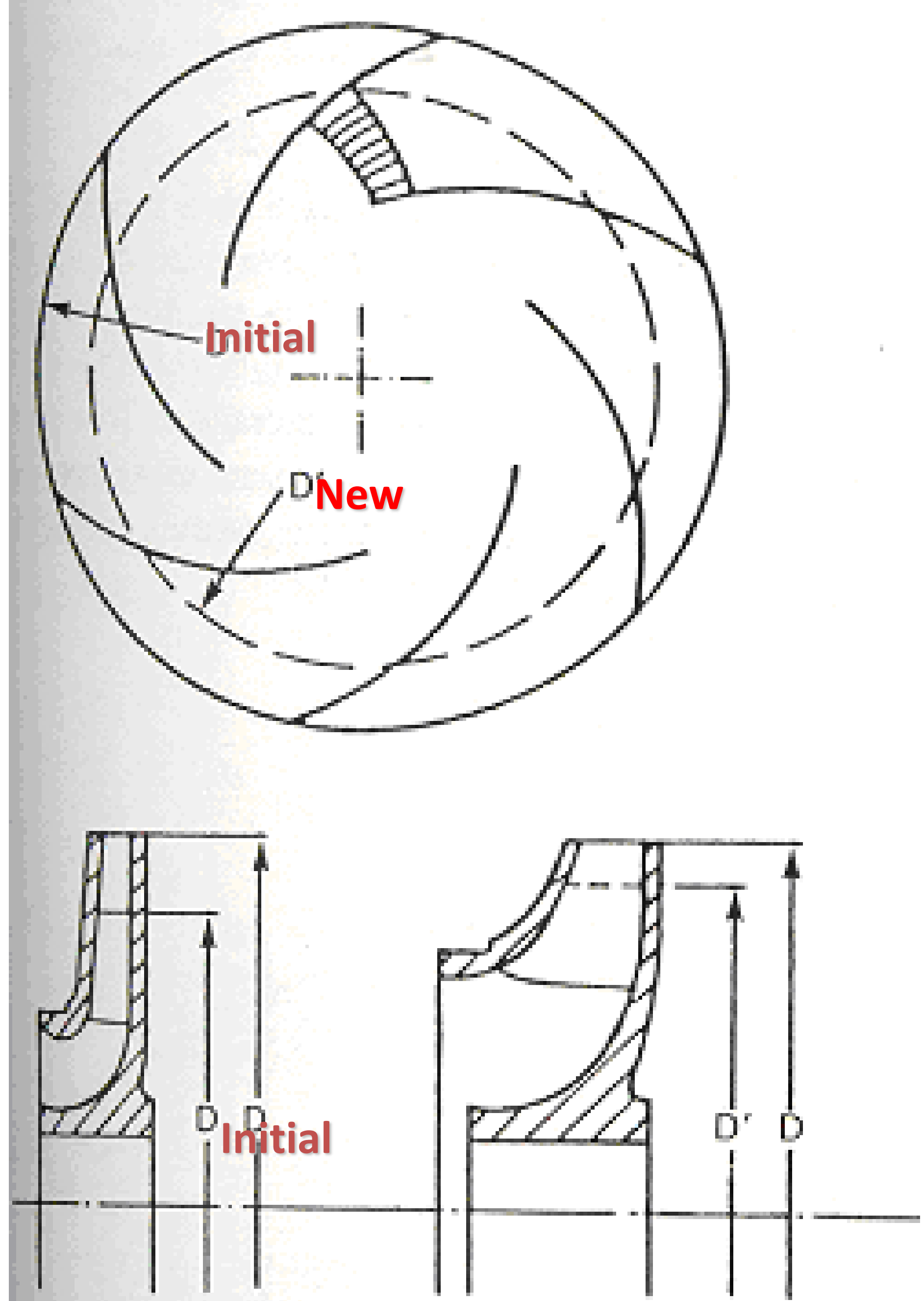


FIGURE 13A through C Diameter reduction of radial-flow impeller (gpm × 0.06309 = l/s). D is measured in inches (centimeters)

“Trimming ”

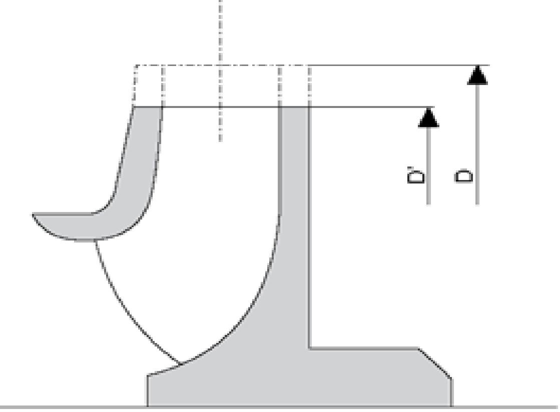
pump impeller to D′ must be done in steps. After each step the modified impeller should be tested . Trimming ends up when the required head and discharge are obtained with a modified impeller of Diam.>D′ .



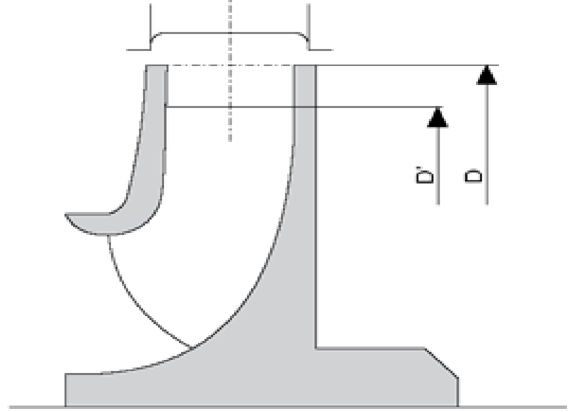
Reduction of pump output by reduction of imp

90

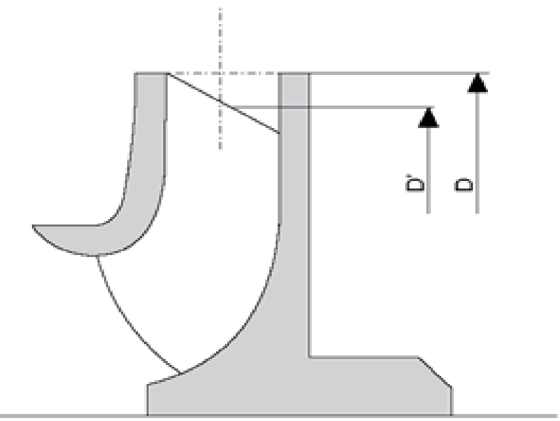
Different types of Impeller Trimming



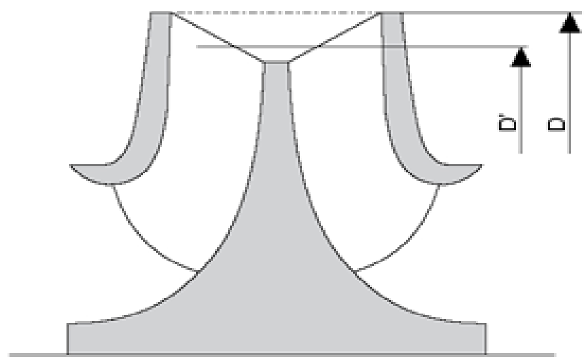
a) Reduction of impeller diameter



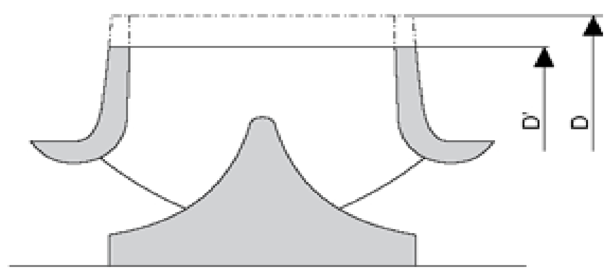
b) Reduction of the blades



c) Oblique reduction of the blades



d) Oblique reduction of the blades



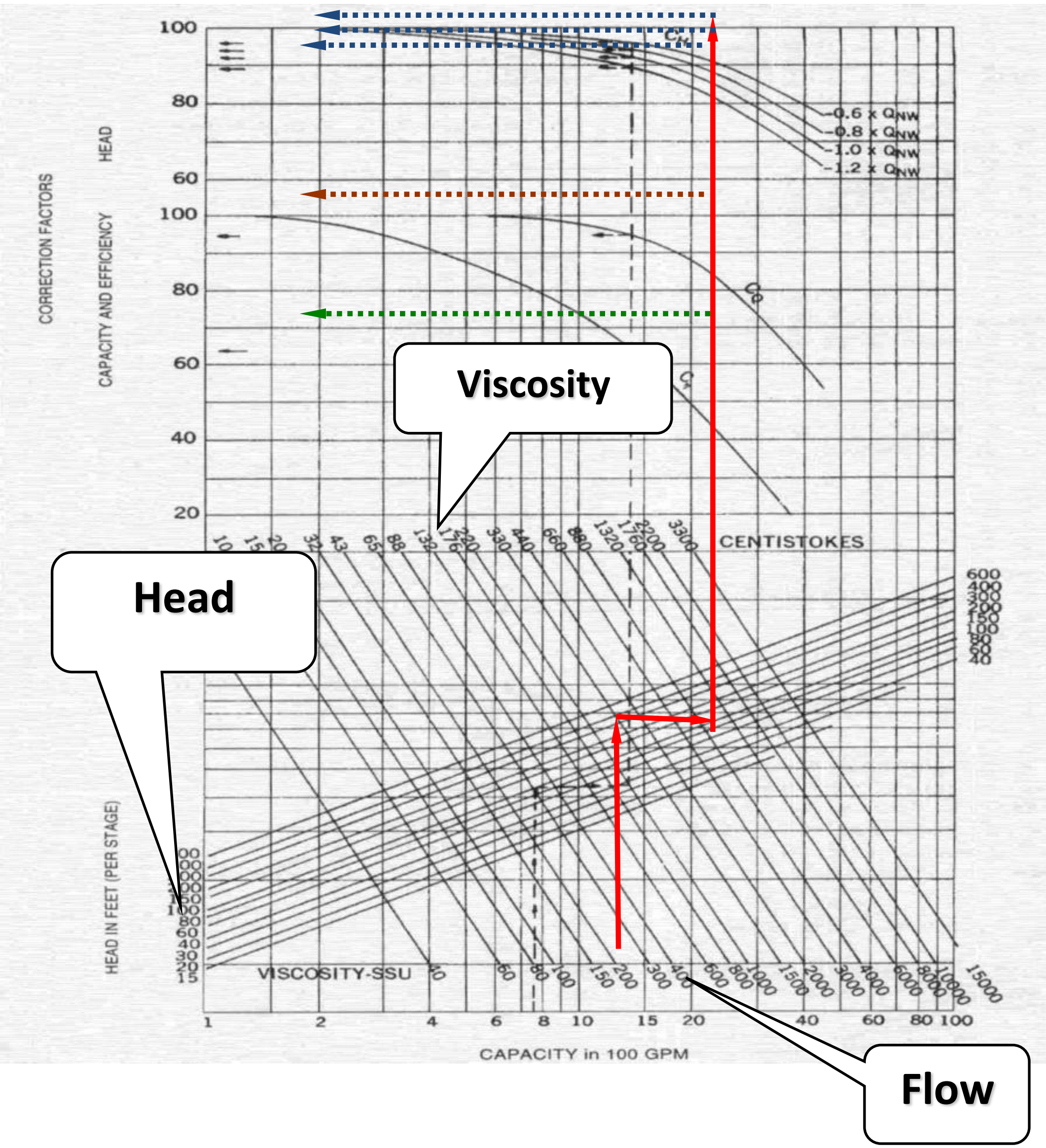
e) Reduction of impeller diameter

Viscosity effects on Centrifugal pump & Viscosity correction

- Pumps' manufacturers test their pumps using water at normal temperature,
- For viscous liquids such as oils, the friction and other losses inside the pump lead to drop in pump's head, discharge and efficiency.
- Viscosity **correction** is necessary when pumping viscous liquid using the nomogram presented By American Hydraulic Institute .

VISCOSITY CORECTION FACTORS

(Courtesy of Hydraulic Institute, 1994 Edition)



. From Q at Bep - move vertically up to the corresponding Head

b) Then move horizontally over to oil Viscosity

c) Then move vertically up to read Coefficients C_n, C_Q and

C_H @ : 0.6 QNW ,

0.8 QNW , 1.QNW

and 1.2 QNW

Poise =0.1 Ns/m².

Stoke =10^{-4} m² /s

From Q at Bep - move vertically up to the corresponding Head

b) Then move horizontally over to oil Viscosity

c)Then move vertically up to read Coefficients C_n, C_Q and C_H @ : 0.6 ,0.8,1,1.2QBep ,

Then Calclate: Qo=CQ.Qw

Ho=CH*Hw

ηo=Cη*ηw

This is applied for one stage in a Multi-Stage C.P.

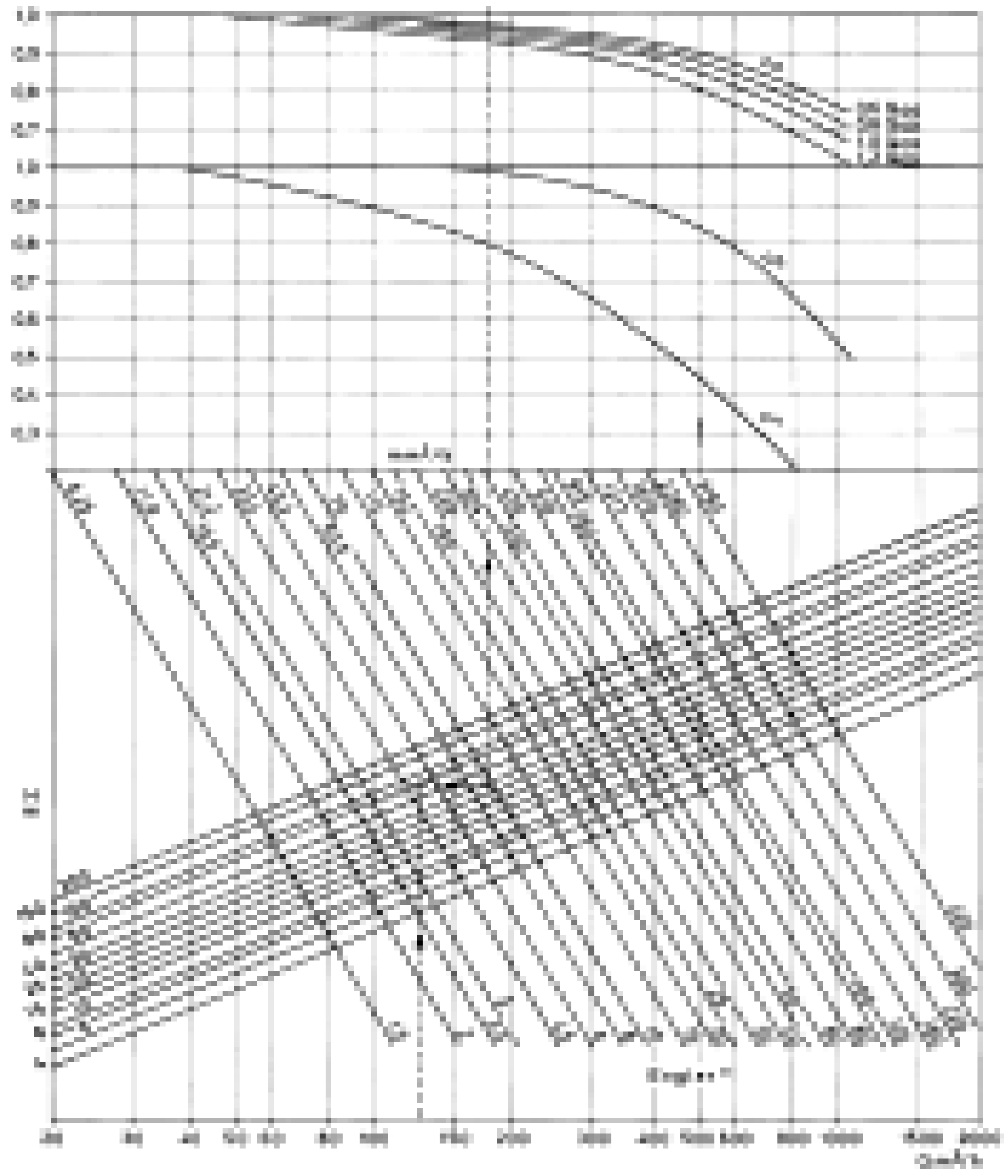


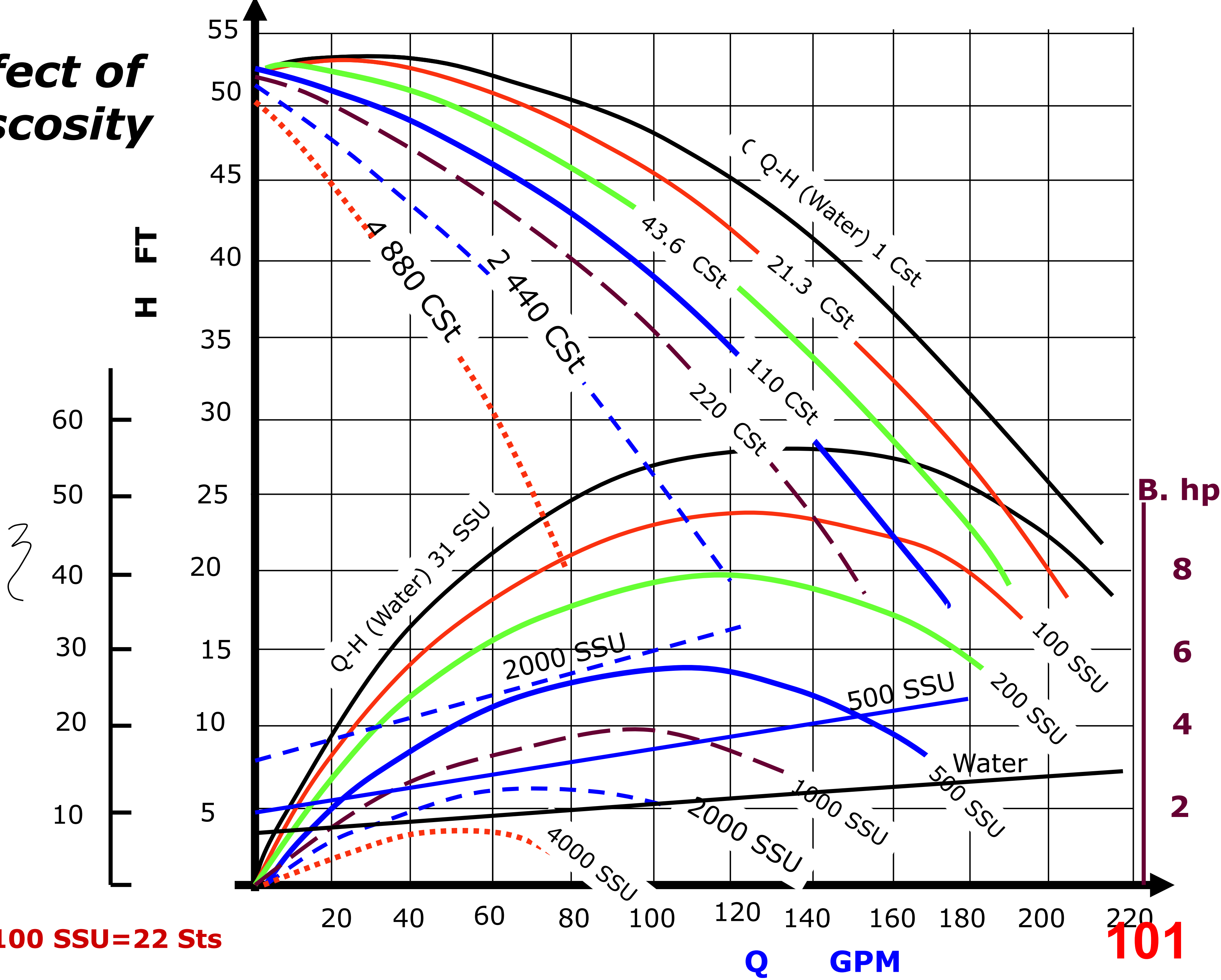
Fig. 2.36
Correction values for pump characteristics when pumping viscous liquids. Hydraulic Institute, 14th edition, 1983.

TABLE 3 Effect of viscosity on performance of a typical centrifugal pump operating at best efficiency point

Viscosity SSU (cSt)	Capacity gpm (m^3/h)	Total head ft (m)	Efficiency %	Brake power bhp (kW)[a]
Nil	3000 (681)	300 (91)	85	241 (180)
500 (110)	3000 (681)	291 (89)	71	279 (208)
2,000 (440)	2900 (658)	279 (85)	59	312 (233)
5,000 (1100)	2670 (606)	264 (80)	43	373 (278)
10,000 (2200)	2340 (531)	243 (74)	31	417 (311)
15,000 (3300)	2100 (477)	228 (69)	23	473 (353)

[a]All values of brake power based on liquid having a specific gravity of 0.90.

Effect of viscosity



100 SSU=22 Sts

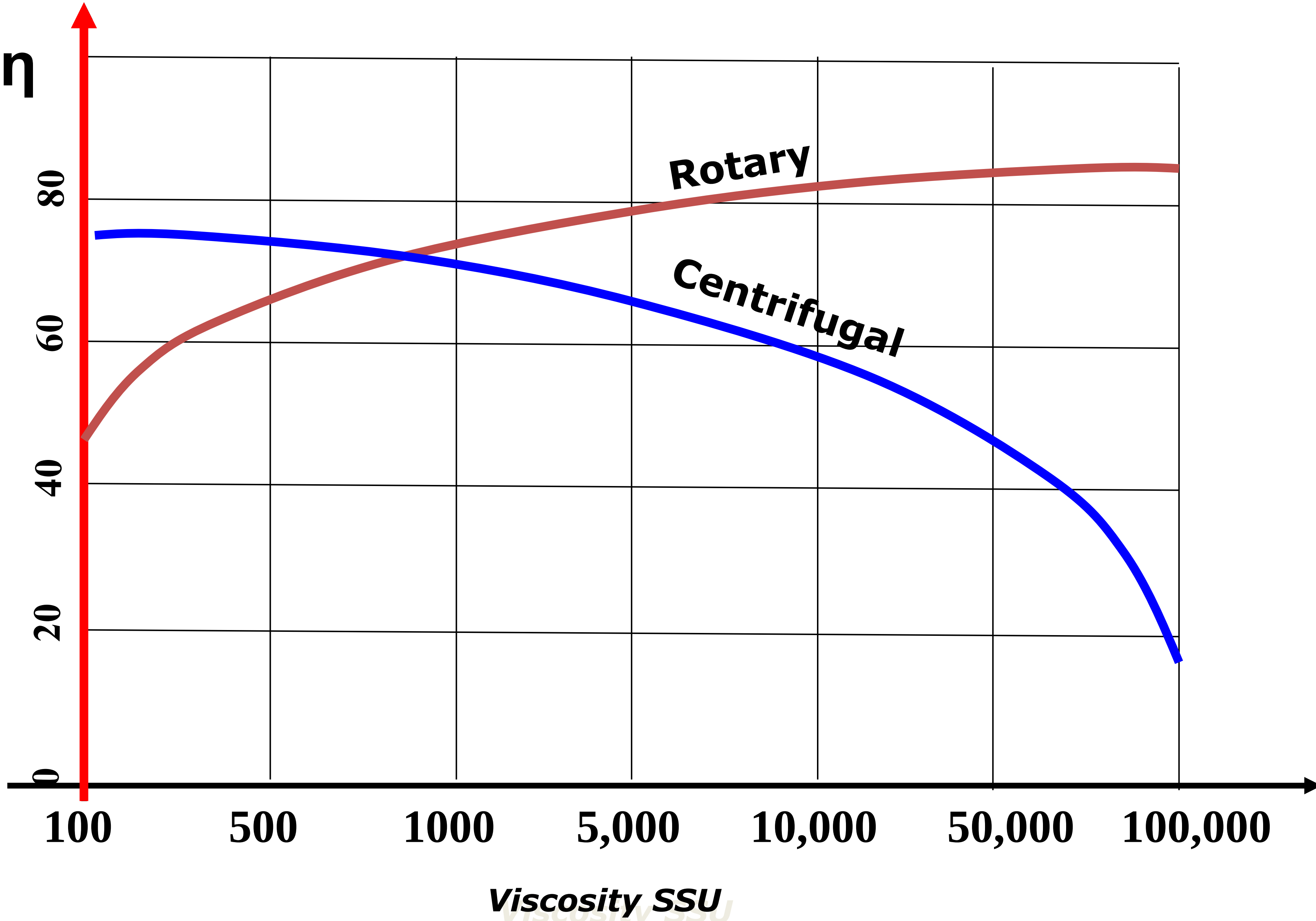
PRACTICAL MAXIMUM VISCOCITY FOR CENTRIFUGAL PUMPS

Where to stop?

If we say that after a pump efficiency is *reduce it to its half* as a limiting rule, then from the chart it follows that:

The *practical* maximum viscosity limit for centrifugal pumps is approximately 500 centistokes

Note: POSITIVE DISPLACEMENT PUMPS CAN HANDLE OILS OF MUCH HIGHER VISCOSITIES with Better Operating Efficiency


η
0
20
40
60
80
Rotary
Centrifugal
100
500
1000
5,000
10,000
50,000
100,000
Viscosity SSU

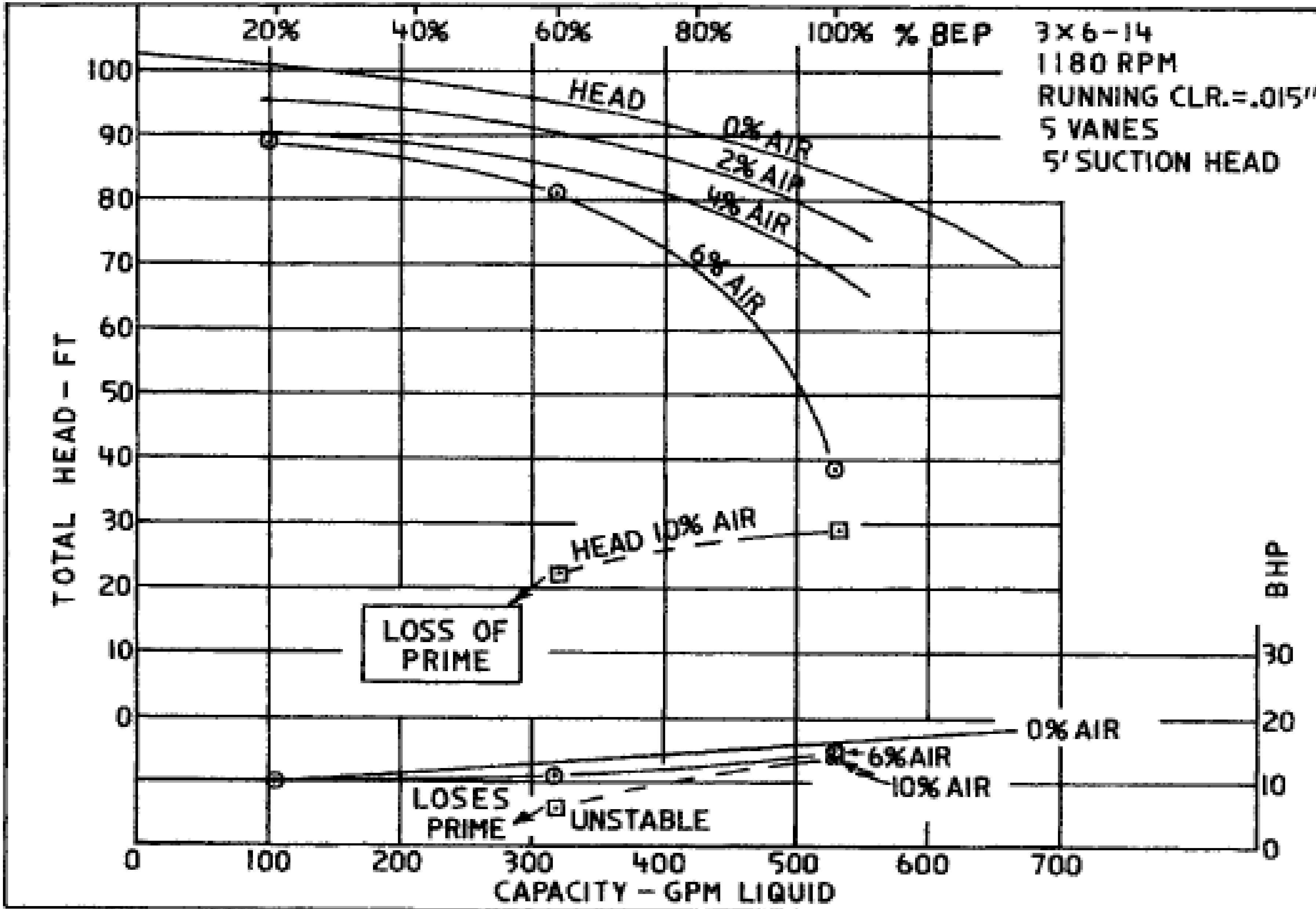


FIGURE 6 General effect of entrained gas on pump performance. (Courtesy ITT / Goulds Pumps, Inc.)

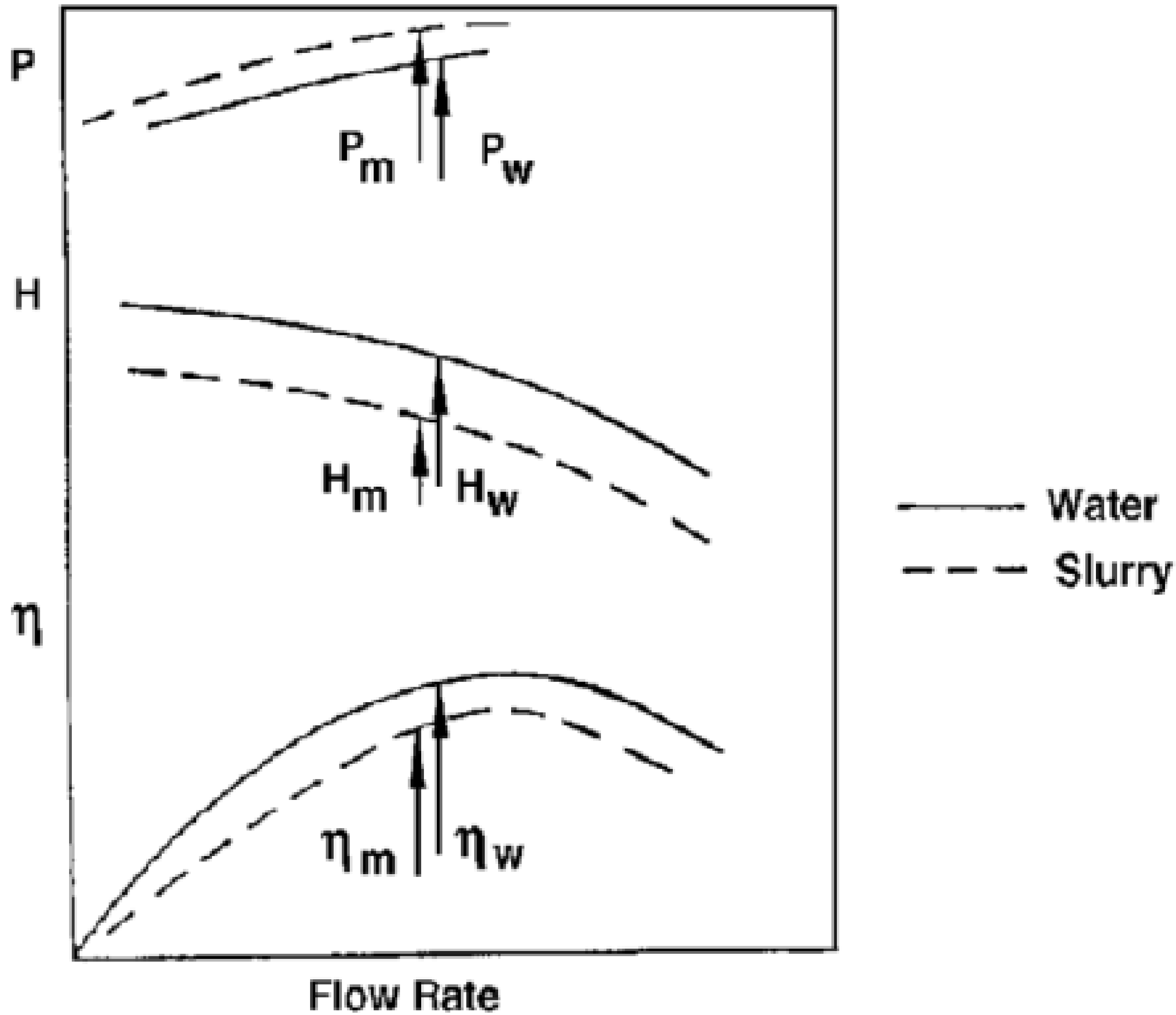


FIGURE 11 Effect of slurry on pump characteristics (schematic)

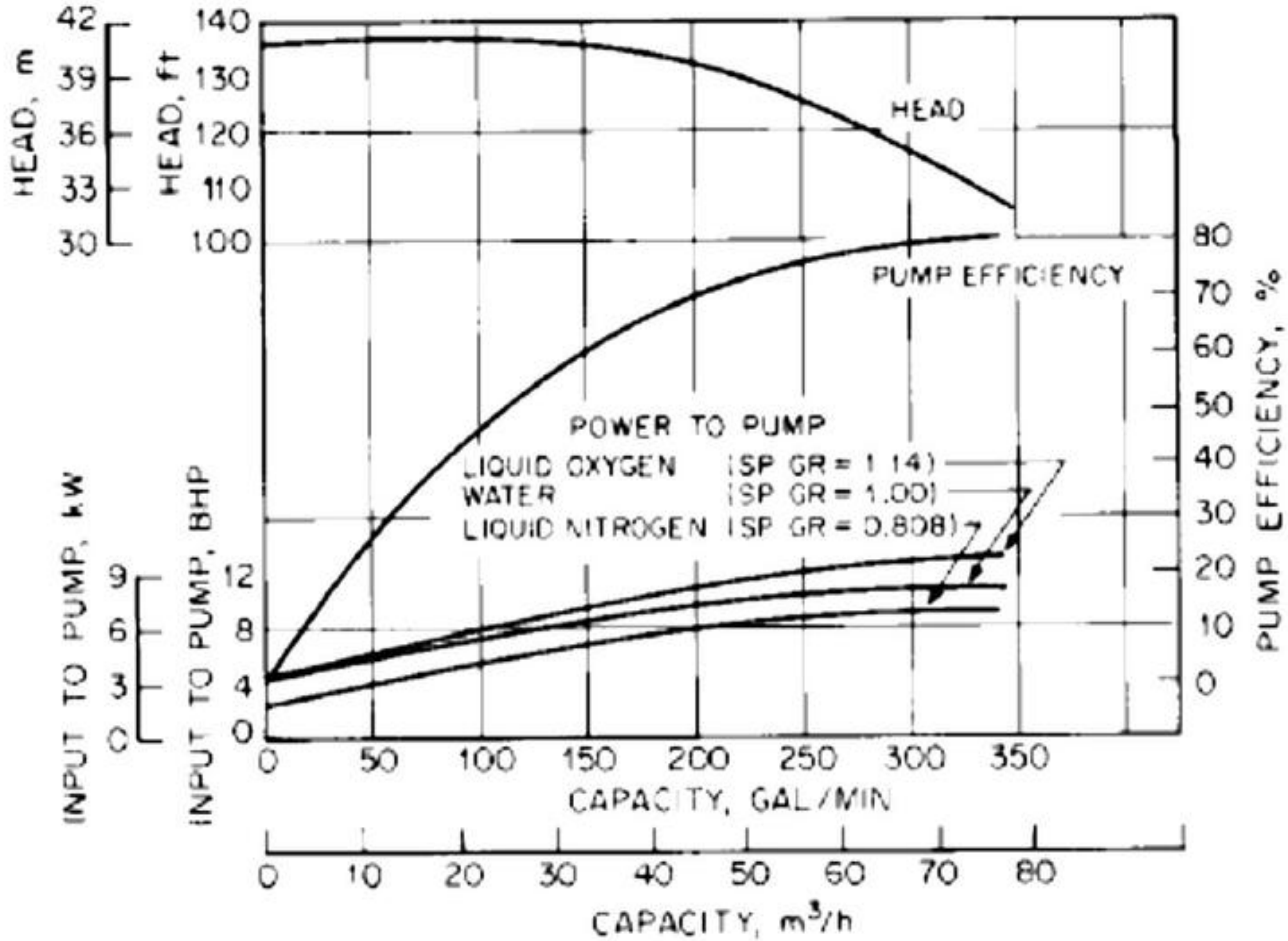
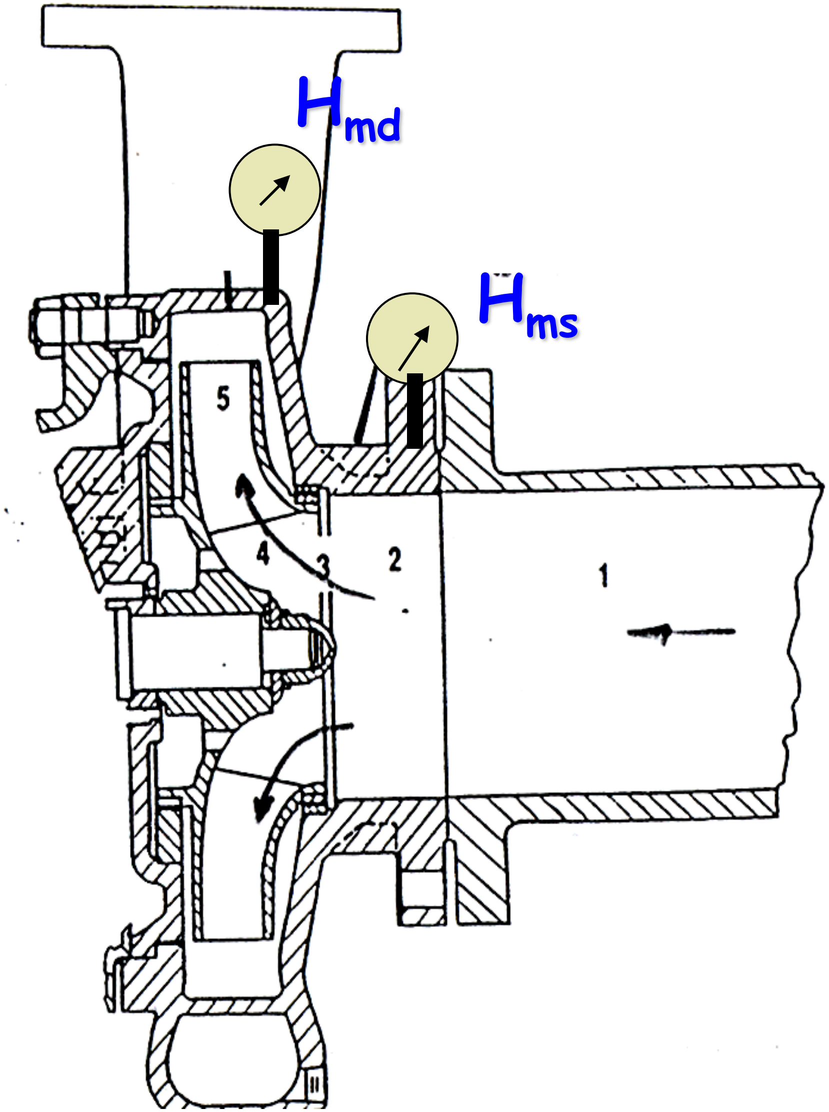
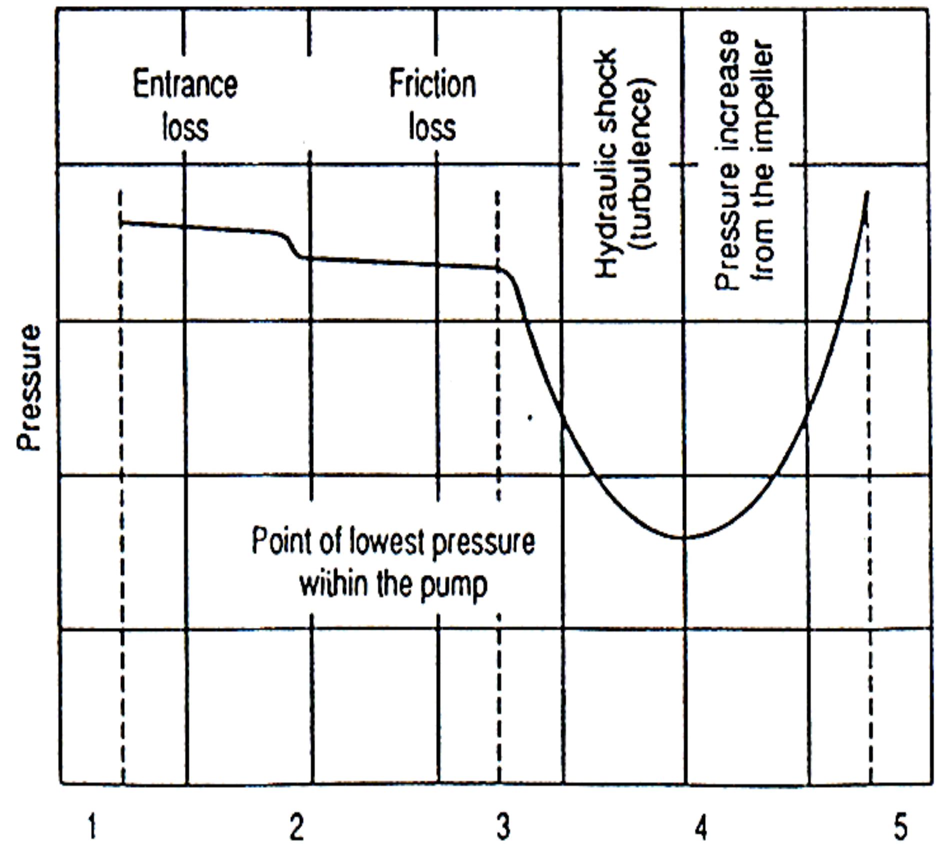


FIGURE 2 Performance characteristics for a refueling pump at 3500 rpm (J. C. Company, Inc.)

Cavitation In Roto-Dynamic Pumps • (C.P.&P.P.)





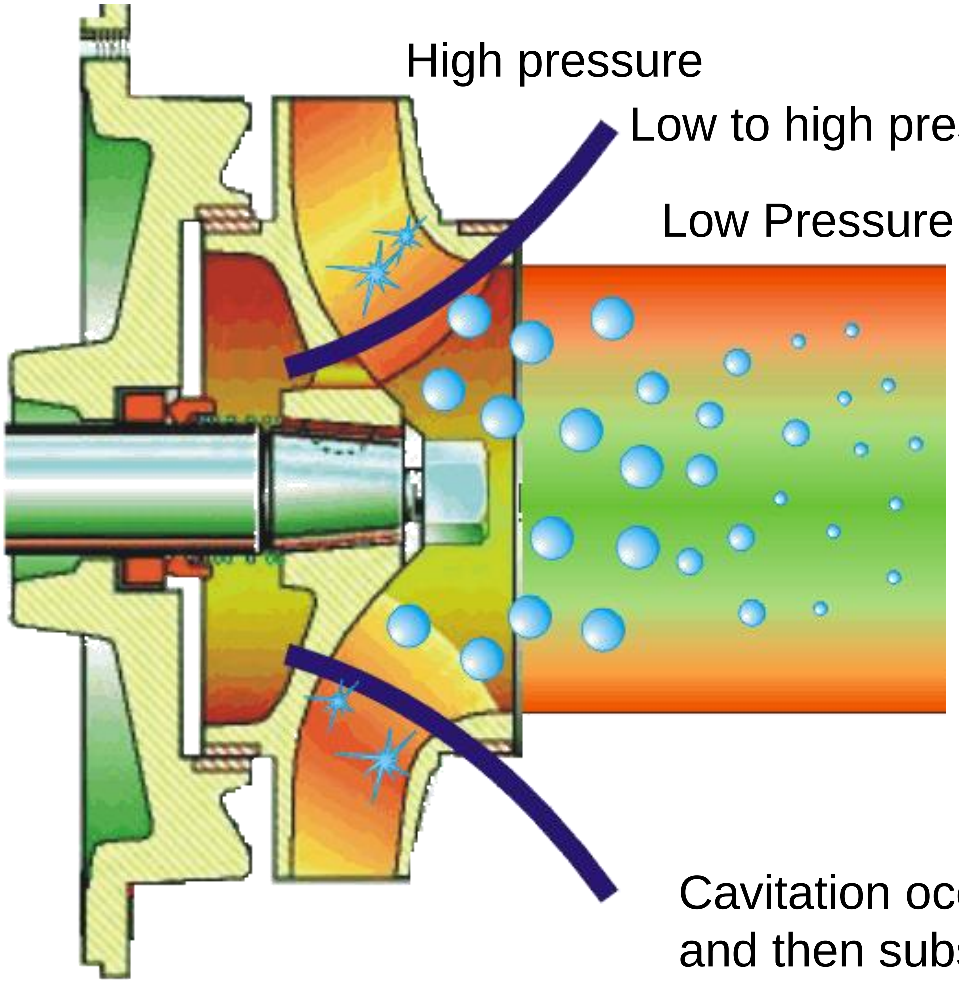
Pressure Distribution within the pump

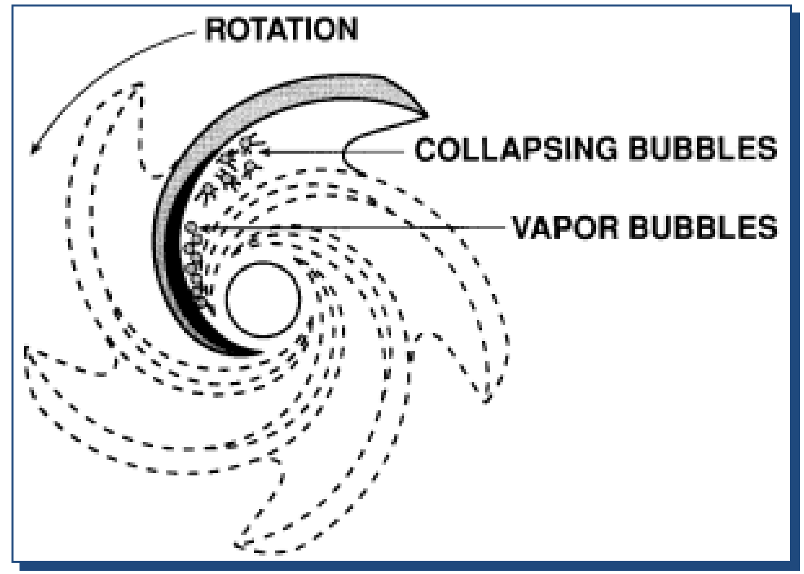
CAVITATION

High pressure

Low to high pressure transition

Low Pressure





Cavitation occurs when vapour bubbles form and then subsequently collapse as they move along the flow path in an impeller.

The minimum head inside suction pipe is at the inlet of the pump & is given by :

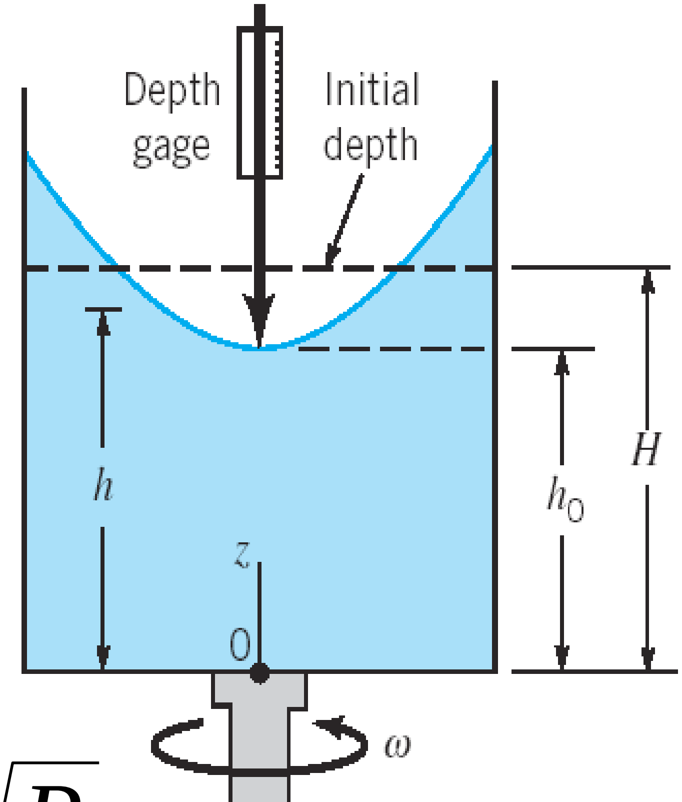
$$H_{ms} = H_{ss} - h_L - \frac{V_s^2}{2.g}$$

In reality, the minimum pressure inside pump does not exactly occurs at the inlet of the pump, but there is an additional pressure drop inside the pump due to the change in flow direction from axial to radial due to very high rotational speed of the impeller (forced vortex). This action leads to an increase in eddy losses and sudden increase in flow velocity followed by reduction in pressure after the inlet of the impeller as shown in figure .

Take **V_s = Flow velocity at impeller eye.**

- $H_{ms} = H_{ss} - H_{l.s.} - v_s^2/2g$

- $H_{min.}$ Inside pump= H_{ms}-X,
- X= Dynamic head depression due to forced vortex near the impeller inlet.
- If $H_{min.} < H_{vap}$, Cavitation Occurs.
- X=Function of (N,Q,H_m...) for a pump
- = Cavitation factor (Segma) * Hm
- Cavitation factor depends on pump Ns,
- For no Cavitation ;
- $H_{ms} - \sigma * H_m > H_{vap.} - H_{atm.}$



$$N_s = \frac{N\sqrt{P_o}}{H^{1.25}}$$

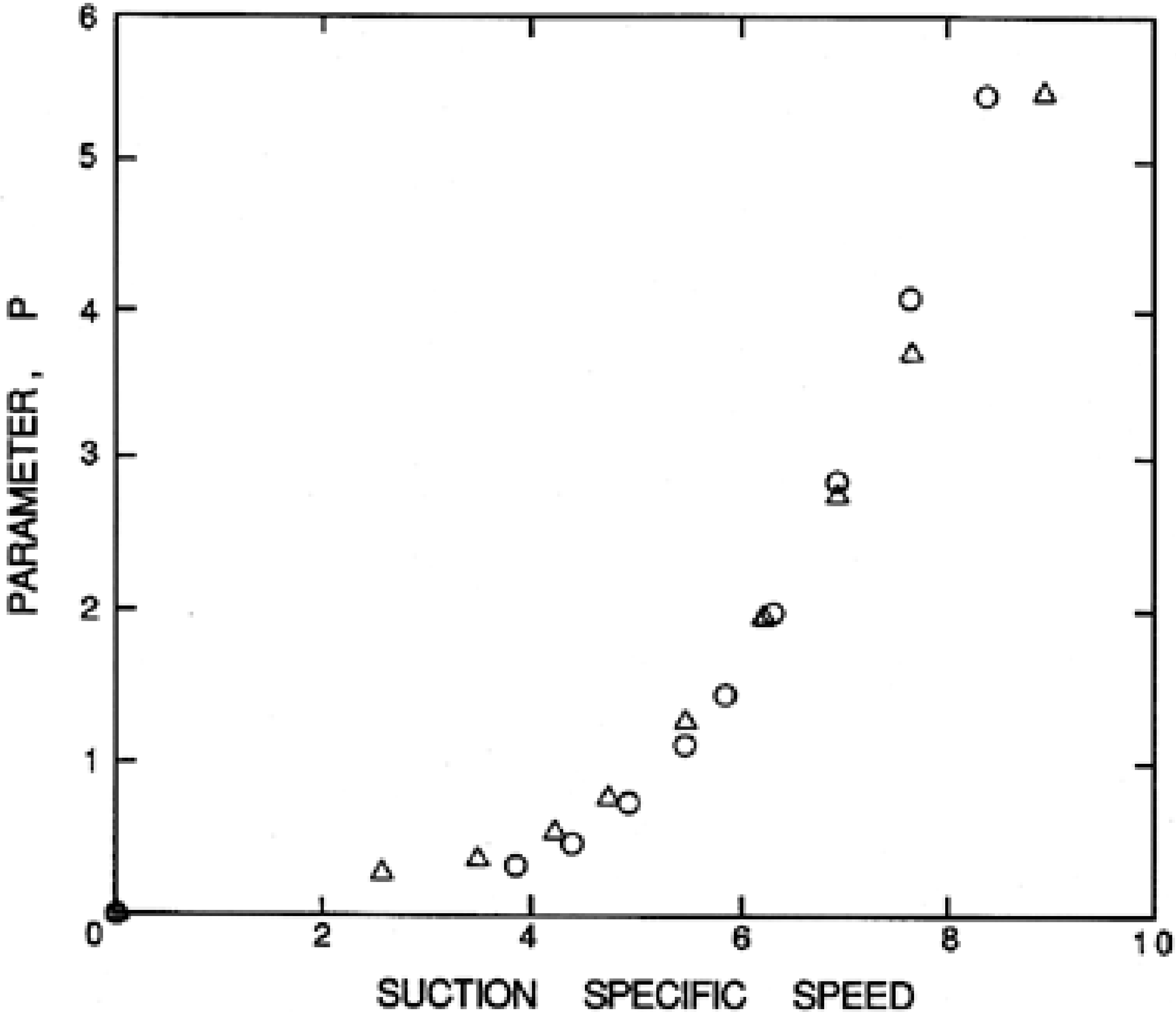


Figure 7.42 Some data on the cavitation head loss parameter, P= ΔH/NPSH, for axial inducer pumps. The two symbols are for two different pumps.

Cavitation begins ,when the pressure inside the pump drops below the vapor pressure of the liquid at the operating temp. , liquid boils up quickly. This occurs at low pressure region just after the impeller inlet. Then rapidly compressed when moved to impeller outlet (high pressure side). Compression of the vapor bubbles produces a small shock wave that affects the impeller surface and pits away the metal creating large eroded areas and subsequent pump failure.

Symptoms of cavitation

Cavitation in pumps can often be detected by a

1-characteristic generated Noise. It sounds like *gravel* inside a concrete mixer due to bubbles generation and Collapse.

2. High Vacuum reading on suction line.

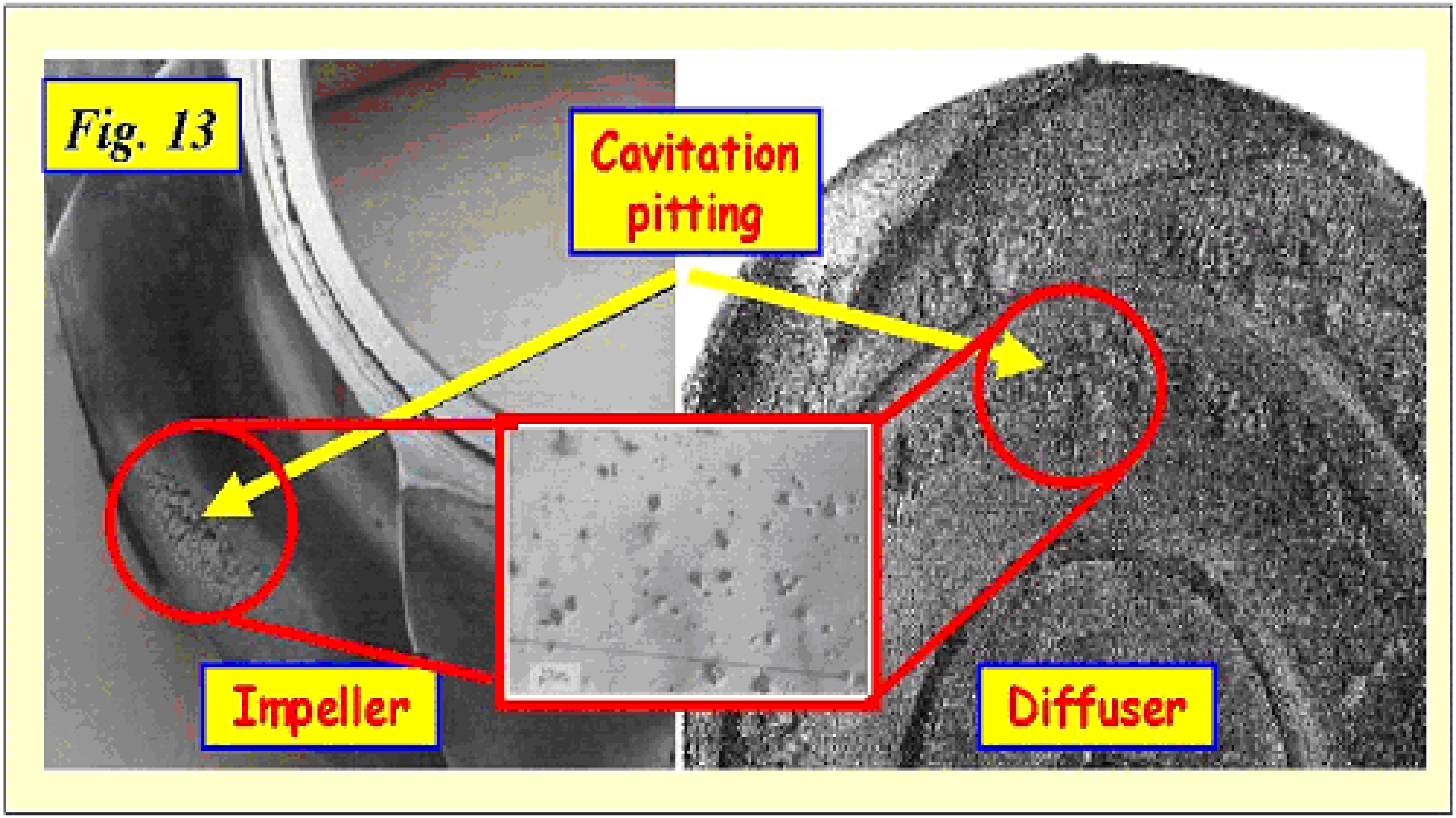
3. Low discharge pressure & low flow

4. Excessive Power consumption .

Cavitation **leads to excessive vibration, fatigue and greatly increased impeller pitting and wear of pump parts , bearing failures , sealing leakage , etc..**

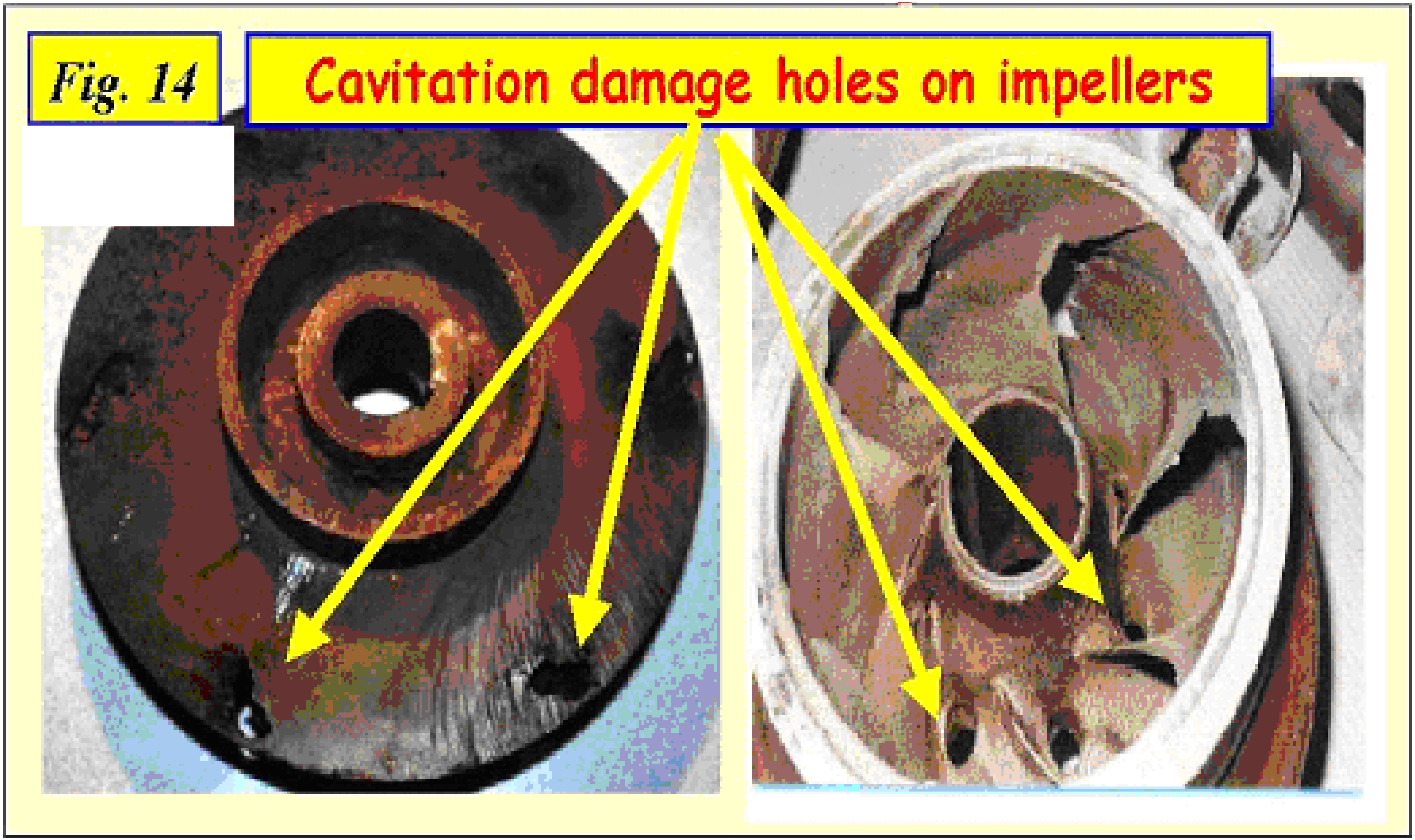
Impeller Damage due to Cavitation

Cavitation pitting



www.cheresources.com

Cavitation Damage



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CAVITATION

Causes

1. Clogged suction pipe
2. Suction line too long
3. Suction line diameter too small
4. Suction lift too high
5. Valve on Suction Line only partially open
6. Discharge pressure too low

Results

1. Reduces pump flow rate and Head .
2. Drop in pump efficiency
3. Pump makes loud chattering noise
4. Future failures of seals on the shaft (Long term
5. Future failures due to metal erosion of impeller (Long term)
6. Shorten Pump Life Time.

To prevent cavitation possible solutions are :

$$Hms - Segma*Hm > Hvap - Hatm.$$

$$Hms + Hatm - Hvap > Segma*Hm$$

or : $NPSH_A > NPSH_R$

The Net-Positive Suction Head Available (NPSHA) is the total suction head, at the impeller eye of the pump minus the vapor pressure head of the pumped liquid.

The term "Net" refers to the actual head at the pump suction flange which should be “Positive” , since some energy is lost in friction prior to the suction.

NPSHR is Net-Positive suction head required by pump manufacturer as stated in pump catalogue.

Factors effecting the $NPSH_a$

The N.P.S.H. available depends on:	Effect on N.P.S.H. available.
1. The friction loss in the pump suction line.	The higher the friction loss, the lower the N.P.S.H. available.
2. The height of the suction tank fluid surface with respect to the pump suction.	The lower the height of the fluid surface, the lower the N.P.S.H. available.
3. The pressure in the suction tank.	This cannot be changed for atmospheric tanks. For tanks that are pressurized, the lower the pressure, the lower the N.P.S.H. available.
4. The atmospheric pressure.	This cannot be changed and depends on the elevation above sea level. The lower the atmospheric pressure, the lower the N.P.S.H. available.
5. Fluid temperature.	An increase in fluid temperature increases the vapor pressure of the fluid which decreases the N.P.S.H. available.

In order to avoid cavitation and guarantee proper operation of the pump, it is desirable to have NPSH available greater than the required NPSH since this allows more flexibility in operation and adds safety towards satisfactory performance.

$$NPSH_{available} > NPSH_{Required} + \text{Safety margin}$$

As a guideline, the NPSH-Available should exceed the NPSH-Required by a minimum of 1.5 m, or be not less than 1.35 times the NPSH-Required,

$$\mathbf{NPSH_{available} > NPSH_{Required} + 1.5m}$$

CAVITATION Remedies

1- Correct selection & installation of pump
2-Increase the pressure at the pump inlet
3-Reduce the rotational speed if possible.
4-Reduce the NPSHR by using an inducer impeller.
5-Minimize the head loss in suction pipe due to friction and fittings to the possible minimum

6. Remove debris from suction line and strainer at suction inlet.
7. Move pump closer to source tank/sump
8. Increase suction line diameter
9. Decrease suction lift requirement
10. Increase discharge pressure
11. Fully open Suction line valve
12. Select larger pump running slower which will have lower Net Positive Suction Head Required (NPSHR)

The pressure at which the liquid vaporizes is known as the vapor pressure and it is specified for a given temperature. If the temperature changes, the vapor pressure changes. Refer to the accompanied table.

Table 1 Water Vapor Pressure vs. temperature {in absolute values }

Temp C	V.pressue KN/m2	Density Kg/m3
15	1.71	999
20	2.36	998
25	3.16	997
30	4.21	996
35	5.61	994
40	7.36	992
45	9.55	990
50	12.31	988
60	19.9	984
70	23.15	978
80	47.77	972
90	70.11	965
100	101.3	958

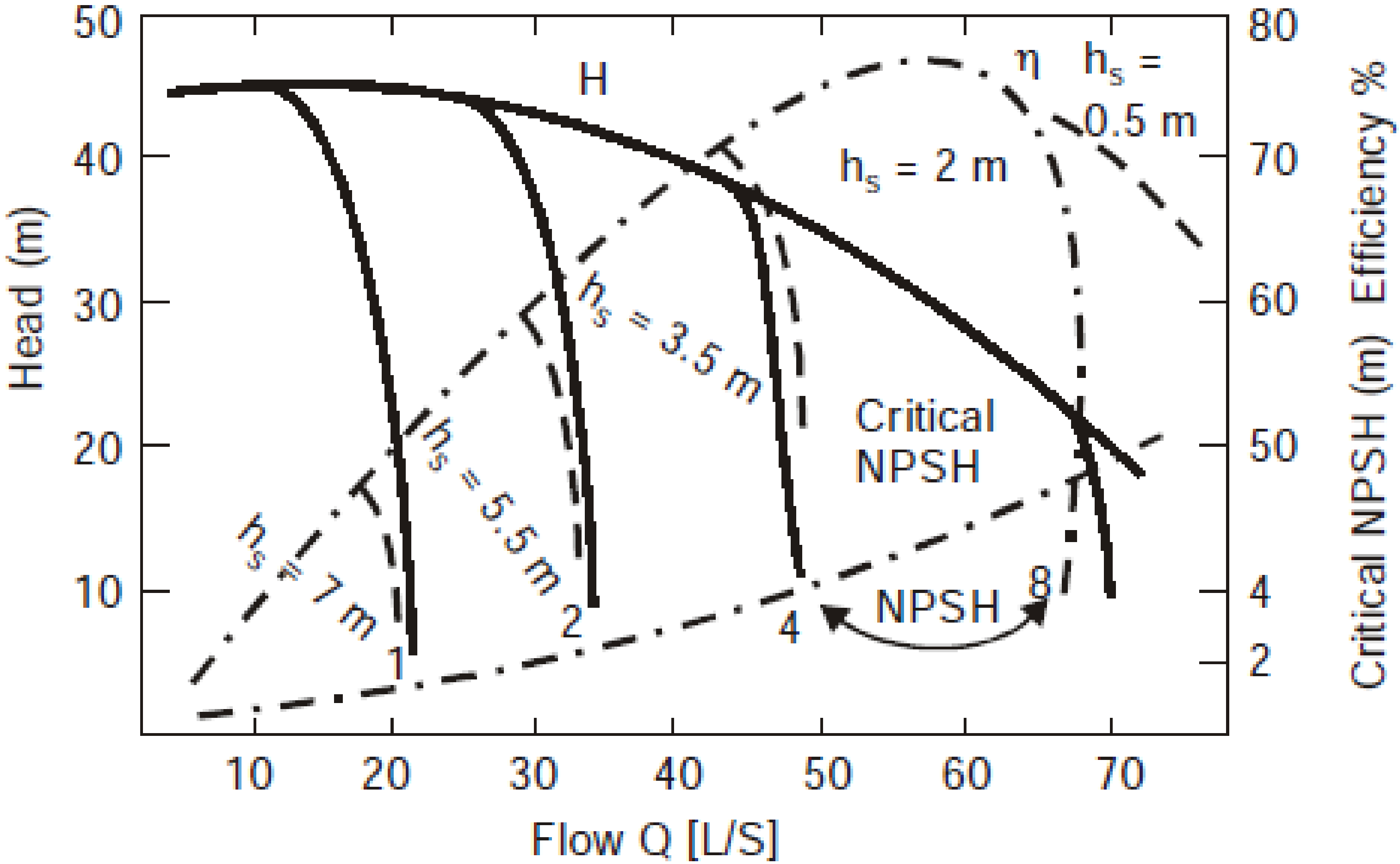
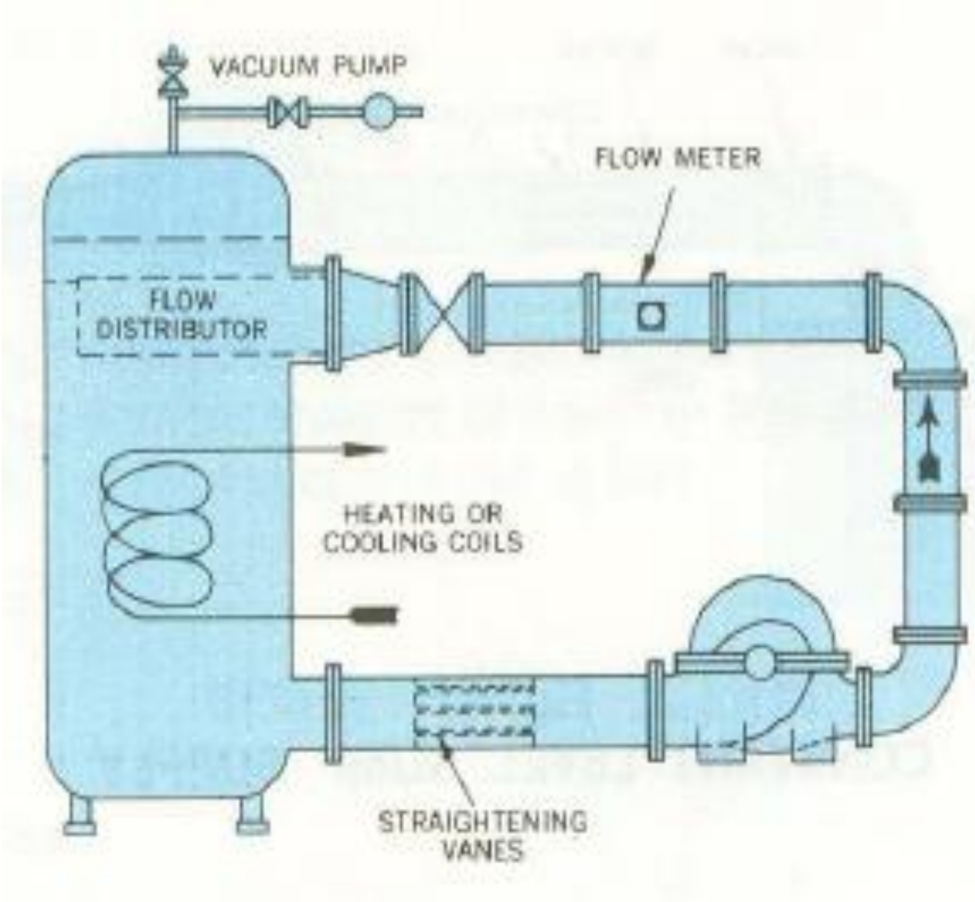


Fig. 9.4. (b) The effect of cavitation on a centrifugal pump performance (effect of suction lift h_s and NPSH)

How the pump manufacturers measure N.P.S.H. required?



The pump manufacturers measure the N.P.S.H. required in a test rig similar to that shown in the corresponding Figure. The system is run in a closed loop where flow, total head and power consumed are measured. In order to provide a low N.P.S.H., a vacuum pump is used to lower the pressure in the suction tank that will provide a low head at the pump suction. The pressure in the suction tank is lowered until a drop of 3% (see next figure) of the total head is measured. When that occurs the N.P.S.H. is calculated and recorded as the N.P.S.H. required for that operating point. The experiment is repeated for many operating points. Heating coils are also used to increase the water temperature thereby increasing the vapor pressure and further lowering the N.P.S.H. as needed.

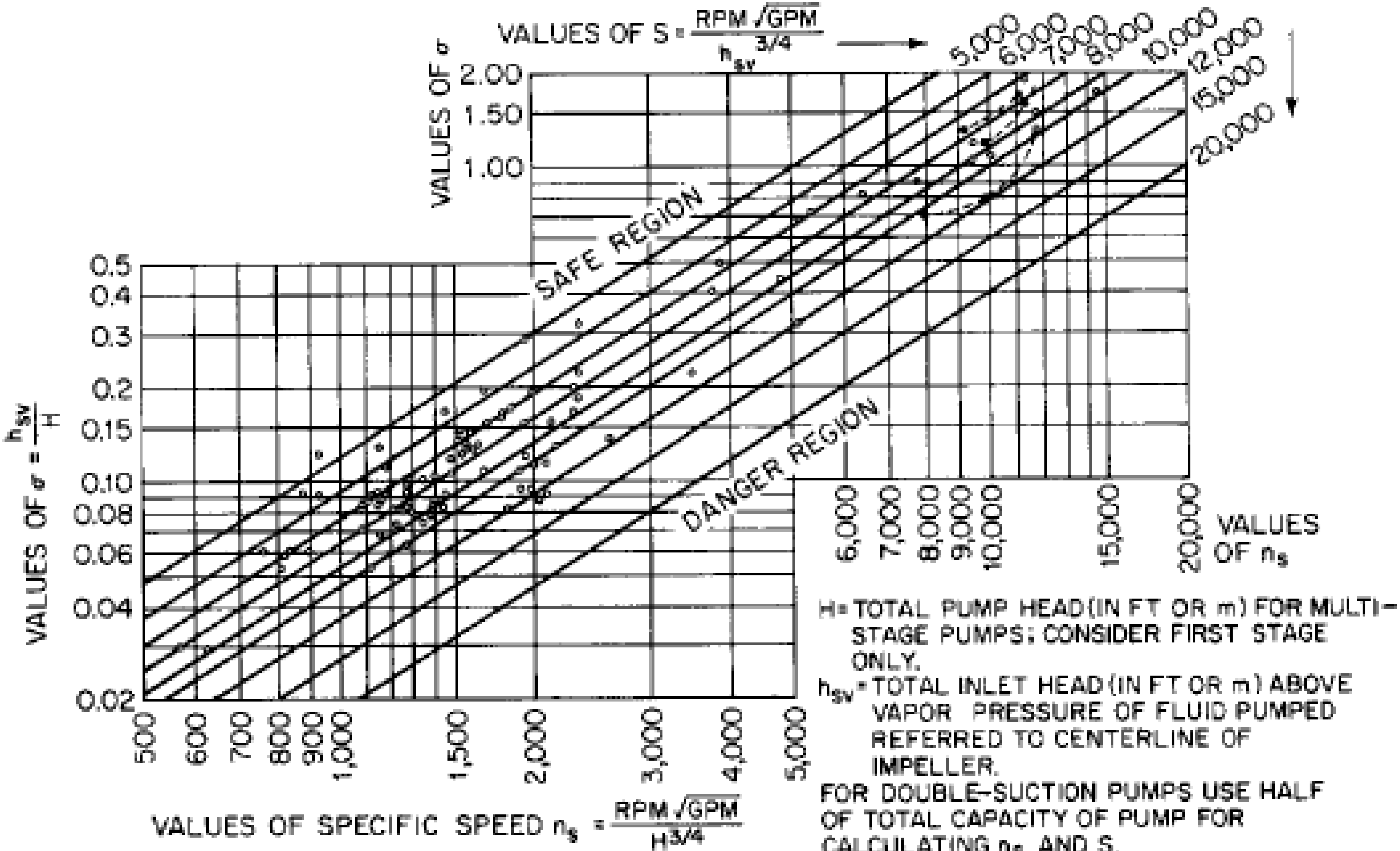


FIGURE 25 Cavitation limits of centrifugal and propeller pumps (Flowserve Corporation)

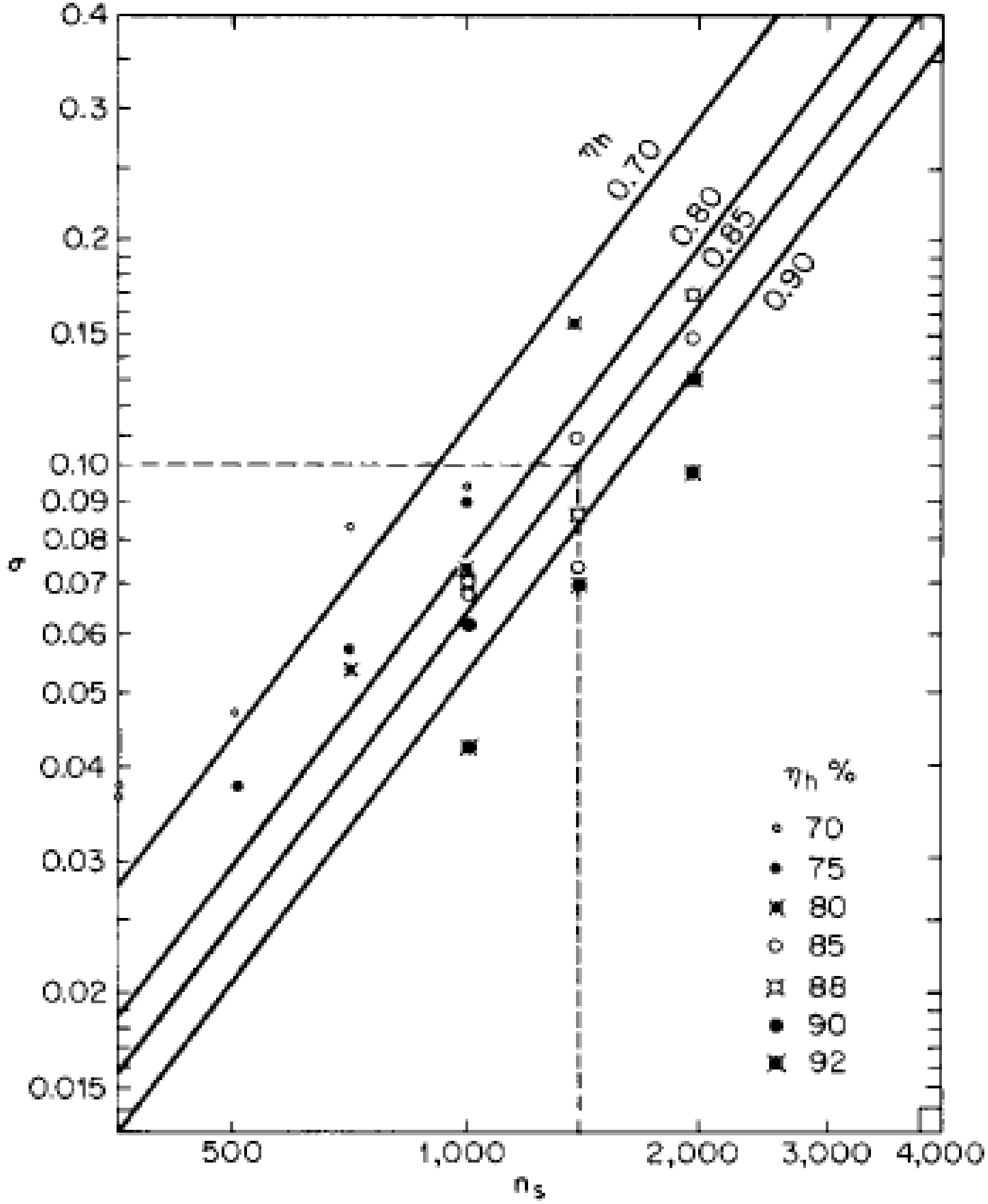


FIGURE 26 Cavitation parameter σ versus specific speed for different efficiencies ($\Omega_s = n_s/2733$)

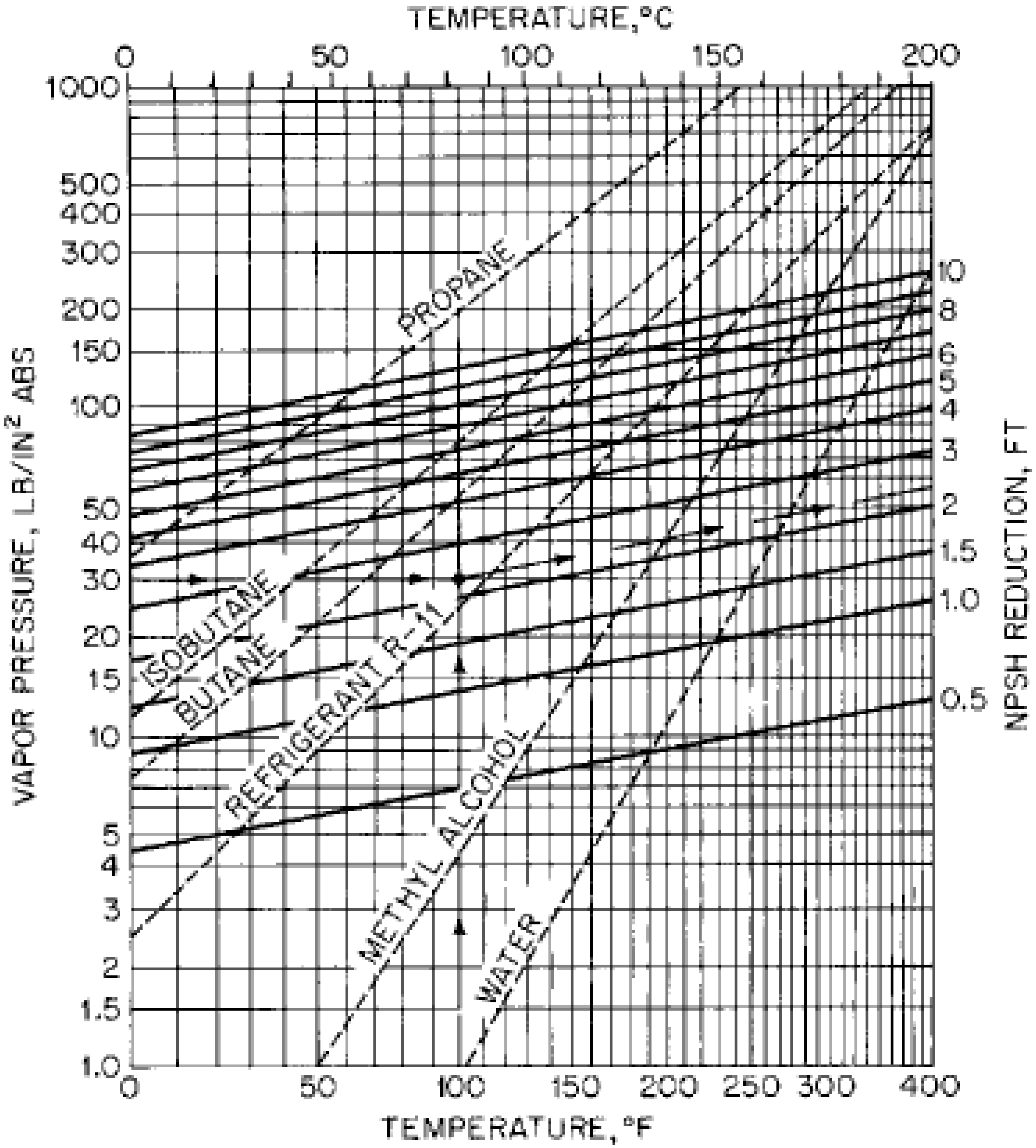


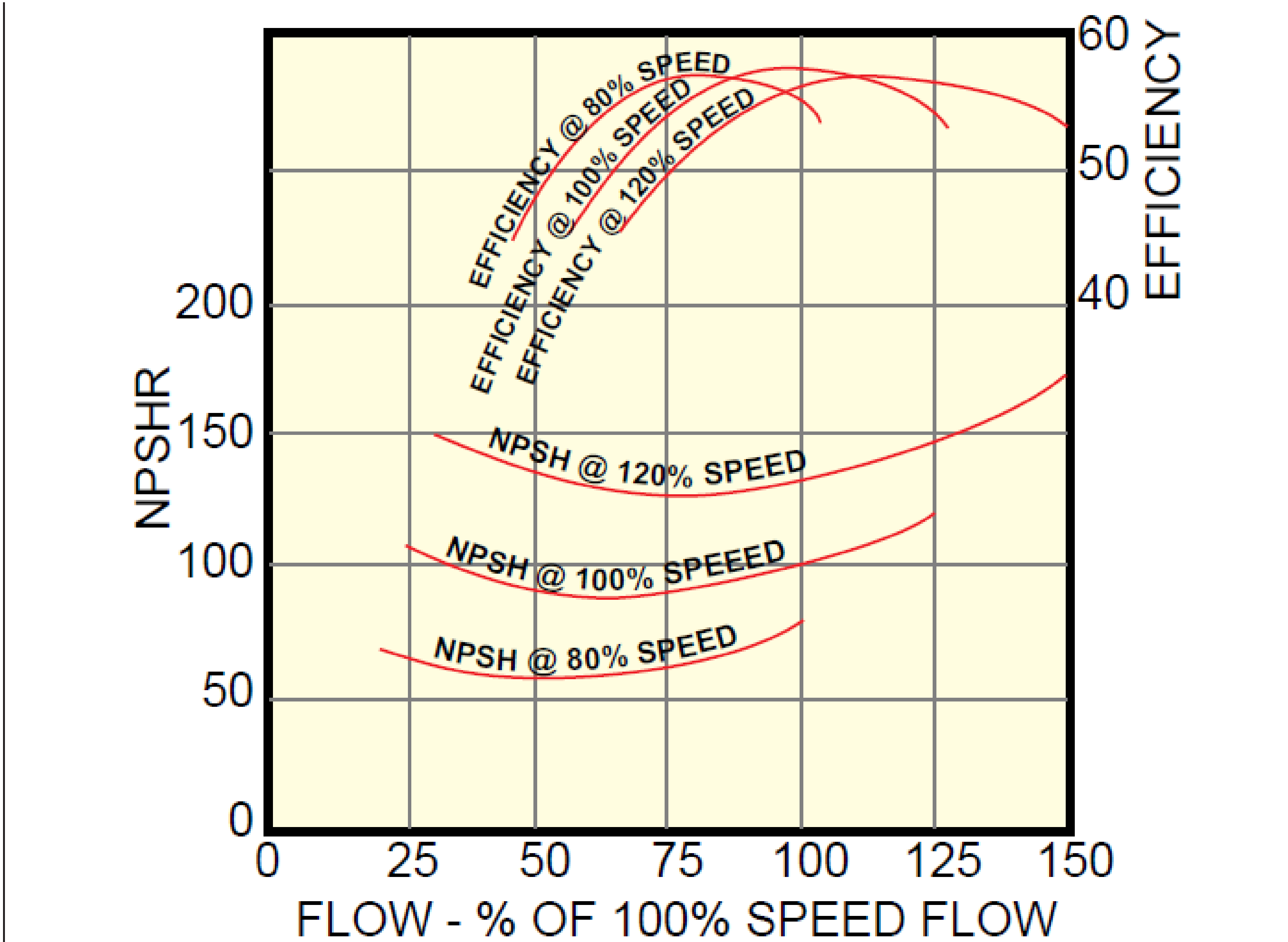
FIGURE 29 *NPSH* reductions for pumps handling liquid hydrocarbons and hot water. This chart has been constructed from test data obtained by using the liquids shown. For applicability to other liquids, refer to the text (ft × 0.3048 = m; lb/in² × 6.895 = kPa). (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 27)

Cavitation erosion resistance of different metals

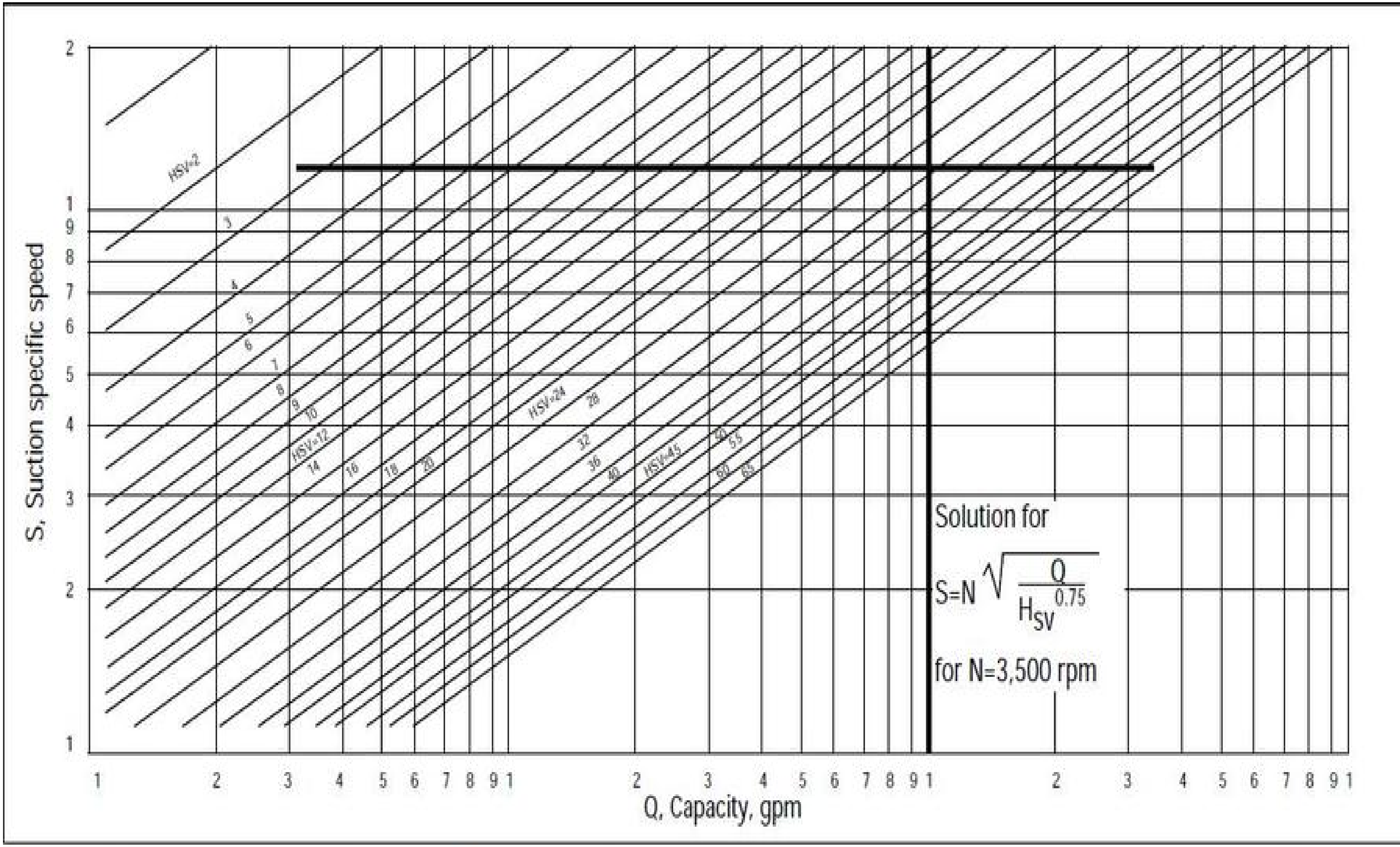
Alloy	Magnetostriction weight loss after 2 h, mg
Rolled stellite[a]	0.6
Welded aluminum bronze	3.2
Cast aluminum bronze	5.8
Welded stainless steel (2 layers, 17 Cr-7 Ni)	6.0
Hot rolled stainless steel (26 Cr-13 Ni)	8.0
Tempered rolled stainless steel (12 Cr)	9.0
Cast stainless steel (18 Cr-8 Ni)	13.0
Cast stainless steel (12 Cr)	20.0
Cast manganese bronze	80.0
Welded mild steel	97.0
Plate steel	98.0
Cast steel	105.0
Aluminum	124.0
Brass	156.0
Cast iron	224.0

[a]Despite the high resistance of this material to cavitation damage, it is not suitable for ordinary use because of its comparatively high cost and the difficulty encountered in machining and grinding.

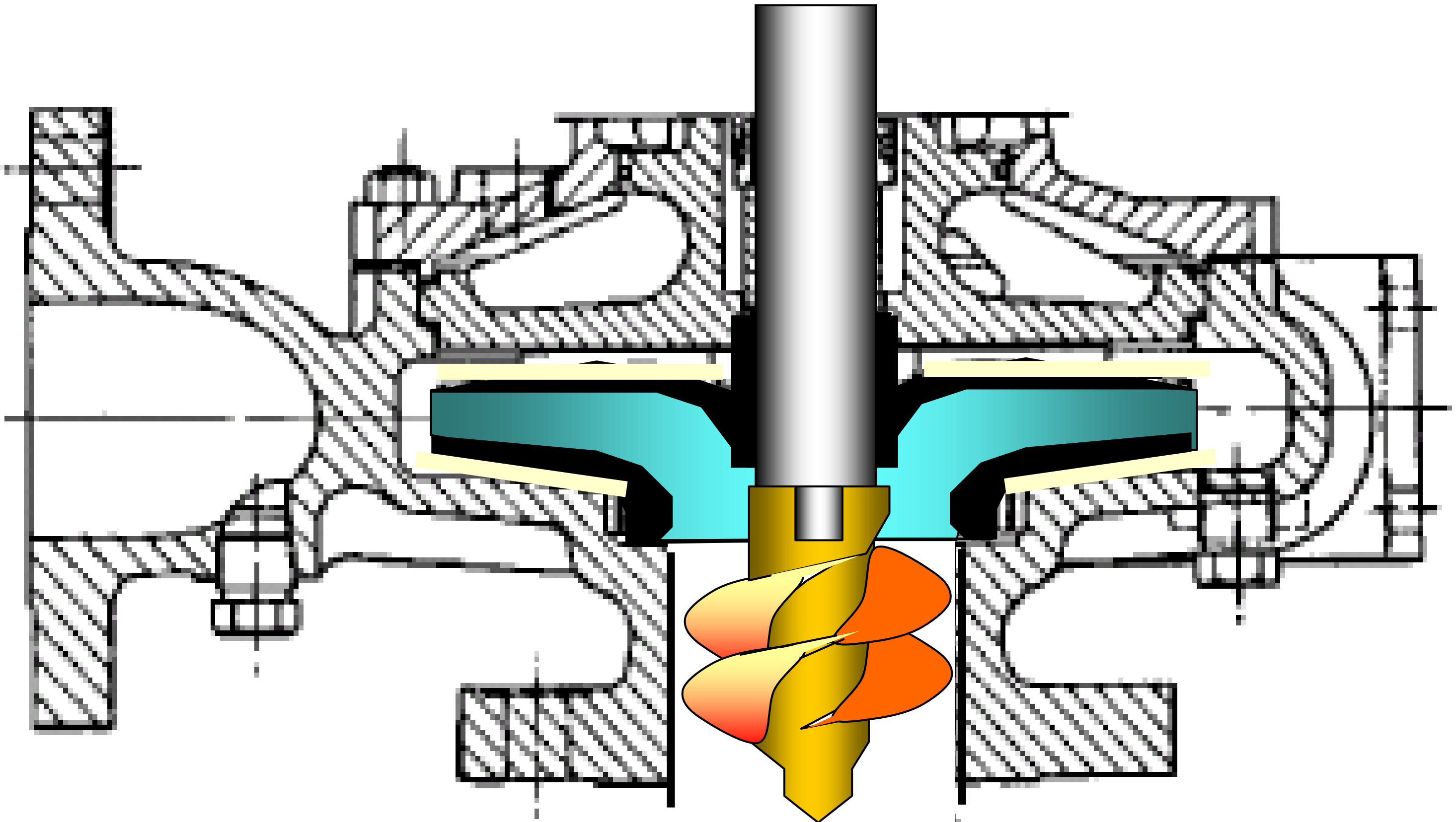
Source: Reference 69.



Centrifugal pump NPSHR and efficiency vs. speed



A plot of suction specific speed (S_s) versus flow in gallons per minute (gpm) for various $NPSH_A$ or $NPSH_R$ at 3,500 rpm. (Single suction pumps. For double suction use 1/2 capacity). $H_{SV} = NPSH_R$ at BEP with maximum impeller diameter.



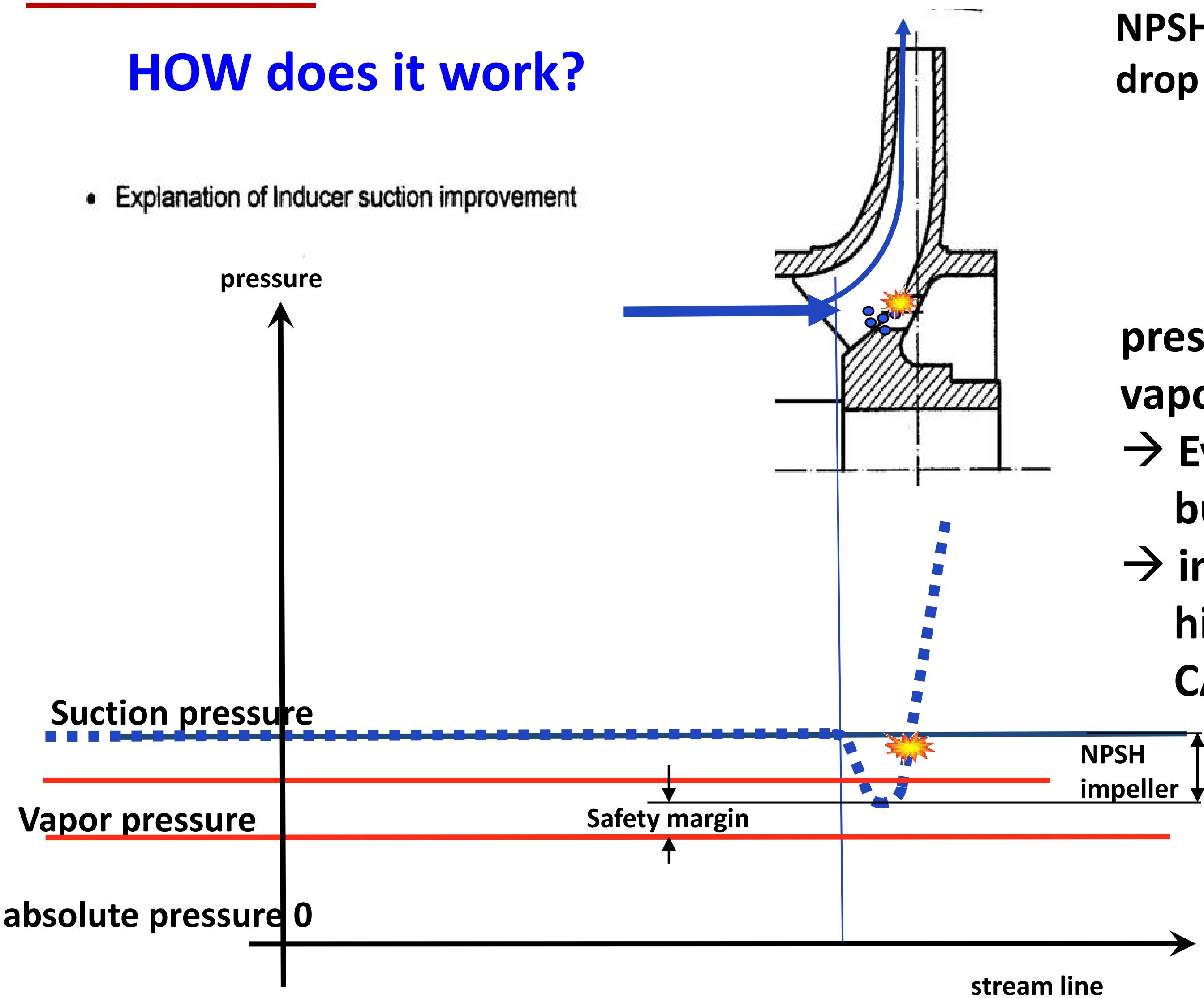
<u>INDUCER</u> **acts as an integrated booster pump, which increases the suction pressure of the impeller inlet to avoid evaporation and also cavitation. It provides a reliable solution to eliminate cavitation problems. It make an essential contribution to raise the operation safety , life time and to reduce the life cycle costs**

INDUCER

HOW does it work?

NPSH impeller = pressure drop at impeller suction eye

- Explanation of Inducer suction improvement



pressure drops below vapor pressure

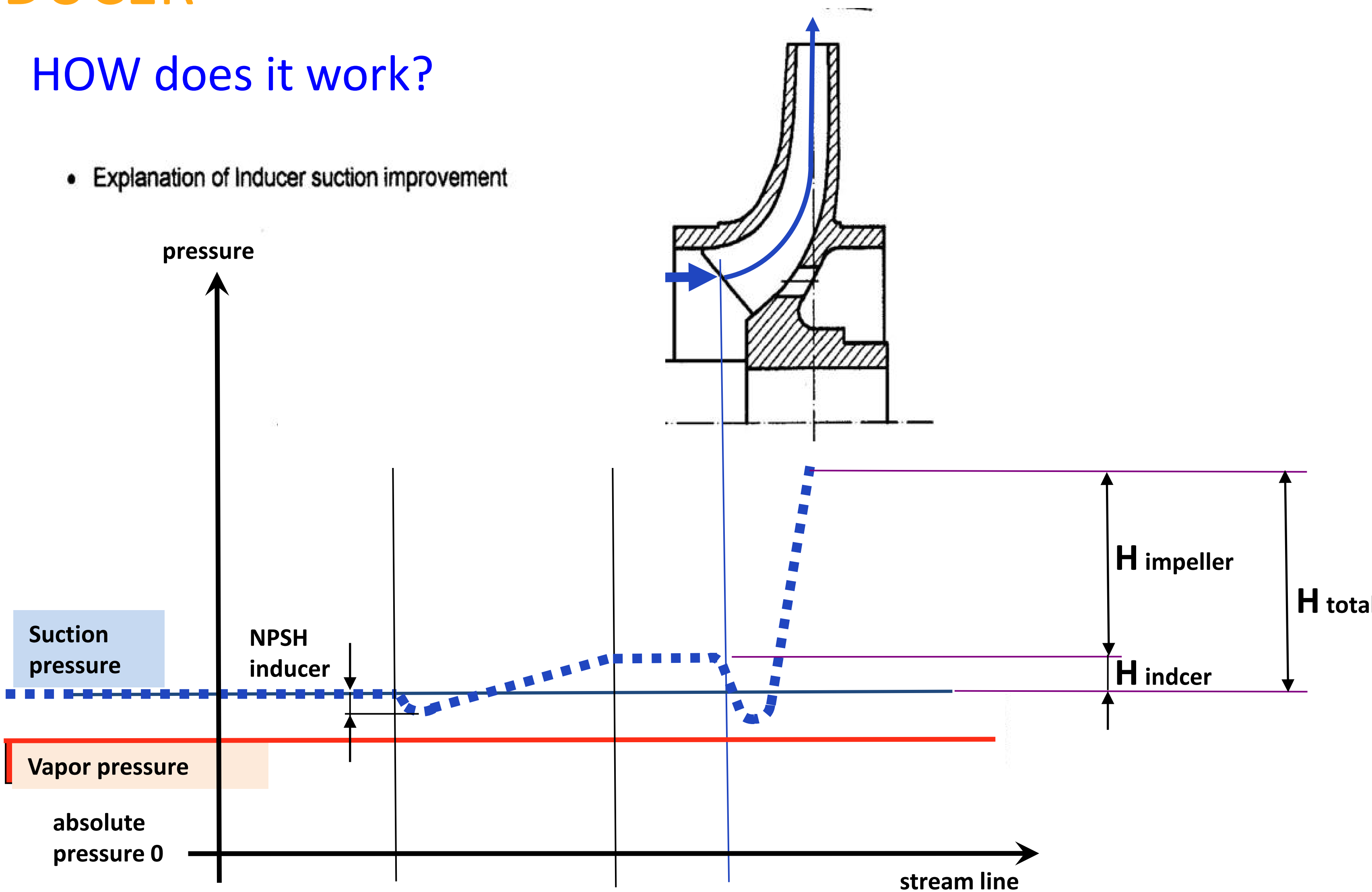
→ Evaporation & Steam bubbles

→ implodes at area of higher pressure = CAVITATION

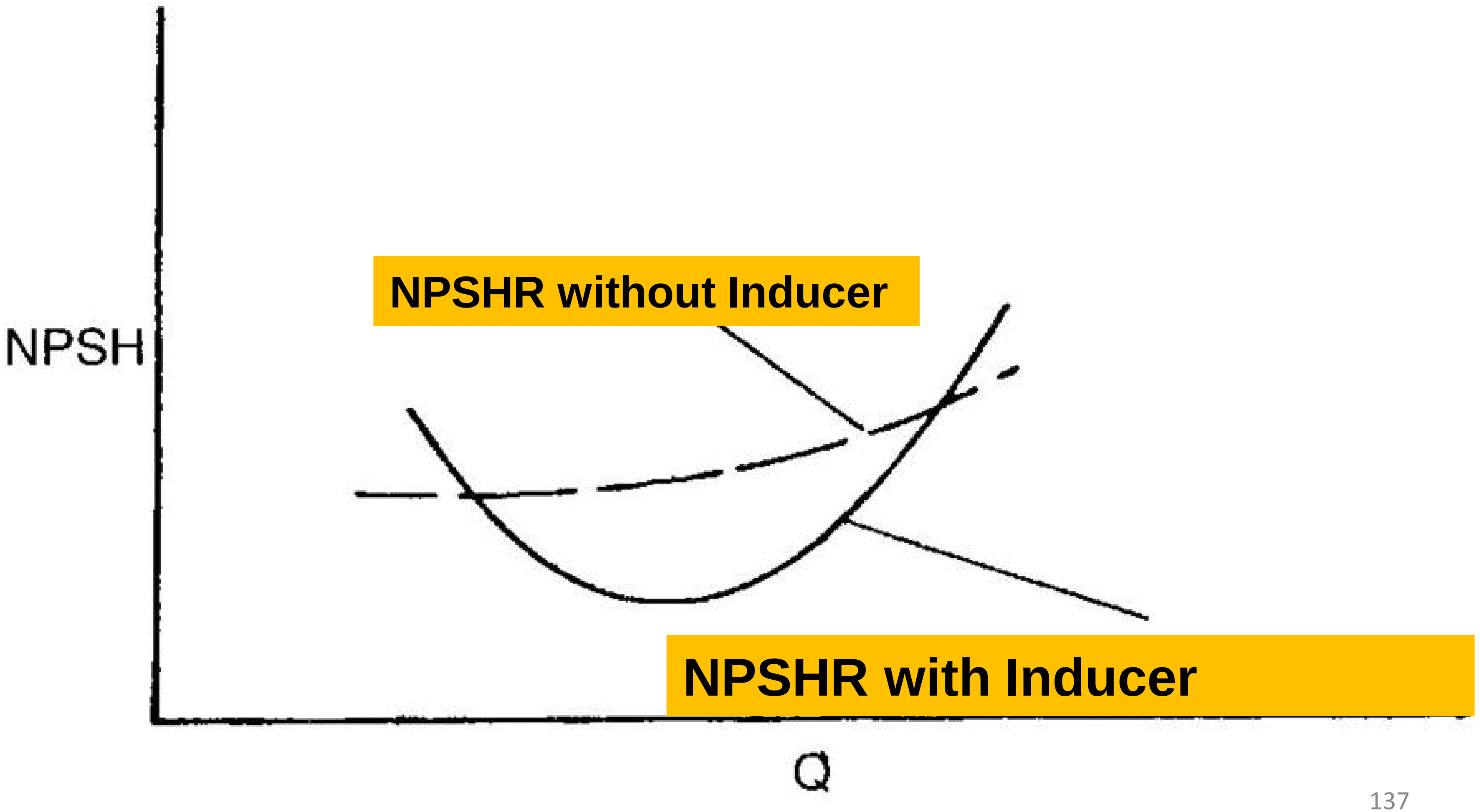
INDUCER

HOW does it work?

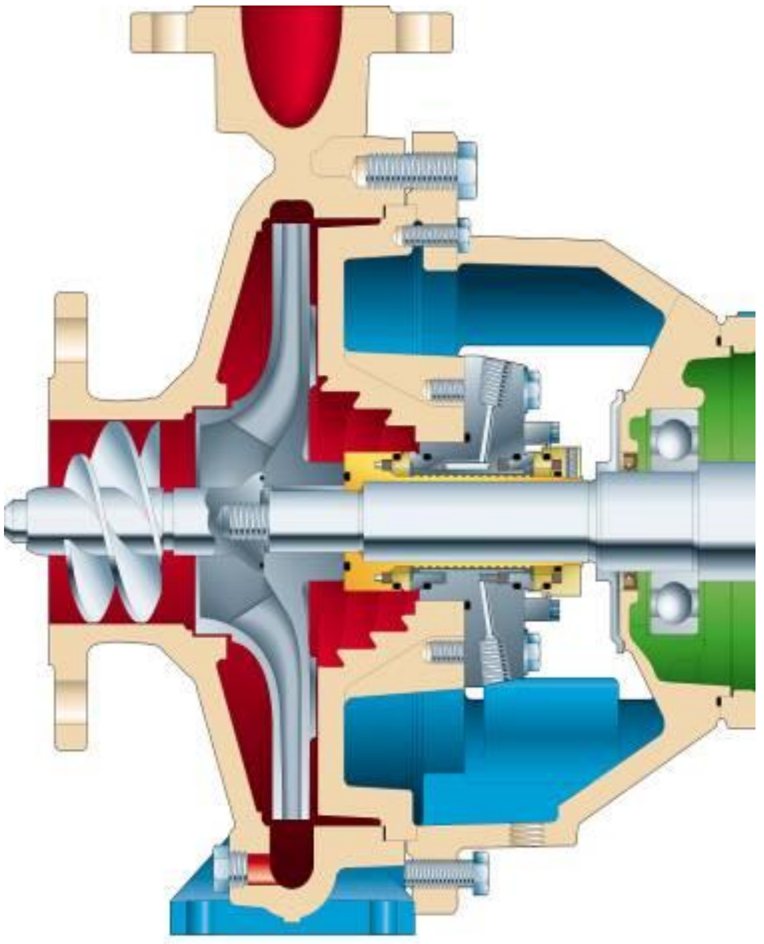
- Explanation of Inducer suction improvement



Conventional Inducer is designed to lower the NPSHR value of the main pump in the range of the duty point, but they only allow a limited operating range of the pump.



INDUCER enables a safe and reliable operation with low NPSH values, the handling of liquids close to the boiling point and the handling of fluids containing entrained gas. The wide operating range allows operation at small capacities without admitting more recirculation and vibrations, and therefore improves the safety in operation of the pump in process applications. These characteristics have a positive effect for the durability of bearing and shaft seal, which leads to a decrease of the life cycle costs (LCC). By means of the Inducer it is possible to replace heavy, expensive, slow running pumps by high speed pumps with better efficiency, smaller dimensions and lower total investment costs, without losing operation safety.



Inducer in a Multi stage-Pump

Inducers can be positioned in front of the first impeller on multistage pumps. The installation then is similar as with single stage pumps.